Chapter 5
Supply of Heat, Cogeneration, and Trigeneration

Complex buildings belong to the municipal sector of the domestic economy, whose share in the demand for final energy carriers is considerable. The demand for final energy carriers (electricity and heat) can be covered by centralized supplies or distributed energy systems and also by both of them jointly.

The heat demand for heating purposes, ventilation, and air-conditioning depends on the external temperature presented in the form of a duration curve characteristic of the given climatic zone. Thanks to the application of hot tap water tanks the demand for heat for the production of hot tap water is stable although the 24 h diagram of the consumption of hot tap water is characterized by considerable fluctuations.

In district heating networks that supply complex buildings, qualitative, quantitative, or both of these methods of control are applied. In the complex buildings themselves, the consumption of heat can also be controlled locally (e.g., pulsatory control by changing the time of operation).

The application of heat pumps in complex buildings depends on local conditions (availability of an adequate bottom source of heat and the efficiency of providing driving energy). Both in the case of the system of centralized supply of heat and distributed energy systems, the application of cogeneration of heat and electricity is more effective due to energy and economic reasons than the separate production of heat and electricity. If district heating systems are situated near system power stations, high efficiency in the production of heat is displayed by the power unit adapted for heat production. Gas as well as gas-and-steam CHP plants are not only ecologically more favorable, but particularly in the case of a combined cycle gas turbine also energetically more effective than CHP plants powered with coal. Small-scale CHP plants fired with gas are generally applied in distributed energy systems. The joint application of a CHP plant and the production of cooling agents lead to "trigeneration" technology. The application of absorption cooler improves the mean annual efficiency of a CHP plant; thanks to the

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partial equalization of the heat loads (increased production of heat in the off-heating season).

Directive 2004/8/EC on the promotion of cogeneration applies the index PES (Primary Energy Savings) as a measure of energy effectivity of cogeneration. In small-scale and micro-cogeneration units $PES > 0$, and in the case of cogeneration units with a power rating equal to 1 MW_e and more, the index PES amounts at least to 10 %, which qualifies these units as high-efficiency cogeneration and entitles them to obtain guarantees of origin concerning electricity production. A comparison of PES concerning various technologies of cogeneration indicates the high effectiveness of gas, as well as gas-and-steam cogeneration units.

5.1 The Municipal Energy Sector as a Part of the Energy System of the Country

Complex buildings constitute some part of the municipal sector of the domestic economy. The share of this sector in the consumption of final energy is considerable, often amounting to about 40 %. Final energy is used for heating, the production of hot tap water, cooling, lighting, catering, and supply of electricity to electrical devices.

The municipal demands for heat are covered by:

- large-scale CHP plants,
- large-scale heating plants,
- local boiler houses,
- small-scale CHP's units,
- stoves.

The energy carriers for heating purposes are:

- hard coal,
- domestic coke,
- wood,
- natural gas,
- fuel oil,
- electricity.

The share of the respective kinds of fuels and electricity varies from country to country. Direct consumption of electricity for heating purposes is rather less applied. The group of heating installations that use electricity also includes heat pumps. The application of heat pumps is connected with a reduction of the temperature in internal heating networks not exceeding 45 °C. This is the same in the production of hot tap water and heating the water in swimming pools.

In the climatic conditions of Northern and Central Europe the application of solar energy for the purposes of heating and production of hot tap water is rather

limited. Usually additional heating systems must be used. For this purpose heat pumps (if they are profitable) or heating boilers are used. Some part of the heat demand for heating purposes may be covered by geothermal water.

The demand for energy required to keep up an adequate thermal comfort in buildings can be reduced, first of all, by applying partitions with more efficient insulation, including windows and doors. A rational architectural solution of the building structures is of great importance. In some situations, buildings have an outer surface nearly 40 % larger than in the case of energy-saving solutions. Sometimes architects design extremely large glassed surfaces, which increase heat losses.

In some countries the difference between the consumption of hot tap water in households supplied from thermal centers (without water meters in the flats or houses) and the consumption of water in households provided with gas or electric heaters is considerable. In households with water meters the consumption of hot tap water is assumed to be half that of in households supplied centrally (without water meters).

Internal installations should be applied in such a configuration so that heat meters can be installed and programmed temperature control may be applied, along with microprocessors. Automation and control applied comprehensively in thermal centers connected with heating networks permit considerable savings in energy required for heating rooms. An essential condition of a rational consumption of heat is the introduction of separate heat meters for room heating and hot tap water. It is possible to apply electrical heating, accumulating hot water in storage tanks which are loaded during the night when the price of electricity is lower. Such a system with microprocessing control permits to keep up a high thermal comfort in rooms over 24 h, provided that the building is adequately insulated. Heating by means of electricity is comfortable because it is easily handled, without causing noise and vibrations and a direct contamination of the environment. Usually, however, it is characterized by considerable thermodynamic imperfection because, besides large exergy losses during the heat transfer from the heater to the heated room, there are also high losses of exergy caused previously in the course of production (in thermal power plants) and transmission of electricity, increasing its costs. Therefore, usually, accumulative electrical heating is used, as has already been mentioned.

Solar collectors are most suitable for the production of hot tap water because they can be exploited all the year round. For heating purposes their application is less effective, because during the heating season their efficiency is lower than in summer. This concerns northern countries. The energy efficiency of a solar collector depends to a large extent on its construction. While choosing its structural parameters, its efficiency ought to be taken into account, as well as the duration curve of the intensity of solar radiation together with the duration curve of the demand for heat.

5.2 Ways to Meet the Demand for Energy Carriers in Complex Buildings

The demand for final energy carriers (electricity, heat, and cooling agents) can be covered in the following ways:

- centralized supplies from electro-energy systems and district heating systems,
- distributed energy systems,
- applying both of these ways (hybrid systems).

The centralized supply of electricity is usually based on domestic or regional electro-energy systems. Depending on the given country either thermal power stations fired with hard coal or lignite (e.g., in Poland), nuclear power stations (e.g., in France), or hydropower stations (e.g., in Norway) dominate. Most often, however, the electro-energy system is in the case of primary energy characterized by the so-called “energy mix” (e.g., Germany, Denmark, Sweden). These latter countries are characterized by the highest share of cogeneration in the production of electricity (e.g., in Denmark about 60 %). In some countries the share of wind power is considerable.

The centralized supply of heat is mainly covered by CHP plants in developed countries. CHP plants are mostly fired with natural gas or hard coal. In the case of gaseous fuels, the thermodynamically most effective way of producing electricity and heat is the application of gas-and-steam cycles. Cogeneration of heat and electricity may also be realized by adapting the power station for the production of heat. This method of heat production has proven to be more effective than traditional CHP units fired with the same kind of fuels (e.g., hard coal). This may, however, be profitable if the distance from the power station to the consumers is reasonable.

Complex buildings are concentrated consumers of heat similarly as municipal consumers are, and are usually supplied from the same district heating systems. In many cases, however, complex buildings are situated too far away for the transmission of heat to be profitable. They may be treated as so-called “energy islands” in which the demand for final energy carriers is covered by distributed energy systems. Distributed cogeneration systems are often integrated with the production of cooling agents. In this way the so-called “trigeneration” systems are realized.

The construction of small-scale CHP plants in the neighborhood of the consumers permits a considerable reduction in both the cost of heating networks and losses of the transported heat. Small-scale CHP plants are constructed, for example, in hospitals, schools, hotels, public buildings, supermarkets, and so on—in short, in complex buildings. They are equipped with piston engines or small gas turbines. Piston engines may be fed with fuel oil or natural gas, as well as biogas.

Exergy losses in the course of transferring heat to heated rooms may be reduced by applying a low-exergy internal installation (floor heating, wall heating, or low-temperature heaters). The application of these installations is, however, only feasible when it simultaneously leads to a reduction of exergy losses in the

production of heat, which may be achieved by means of cogeneration. The application of low-temperature internal installations permits a reduction in the mean temperature of the network water and to increase the production of electricity. A low-temperature heating system permits a reduction in exergy losses in the energy systems because these losses are much lower in a steam boiler than in a water heater [15].

5.3 Supply of Heat

5.3.1 Introduction

Heat is one of the oldest forms of transforming energy applied usefully by humans. Originally, it was obtained by burning wood, and later by combusting coal and other fuels. This form of utilizing the chemical energy of fuels is still applied in our time. Charcoal as a form of smokeless fuel was used already by the Romans, who also invented the first central heating. It was a kind of floor heating in which the flue gases from the furnace situated underground flowed through a system of channels under the ground floor. The water containers above the furnace chamber were the first installations that provided centrally hot tap water.

In the Middle Ages, air heating was applied and furnaces made of stone were used for this purpose. By burning wood the layers of stone were heated up, transferring the accumulated heat to the air inside the room. This heating system was applied, among others, in the castle of the Teutonic Knights at Malbork (Poland). In 1884, a system in which air was heated through masonry walls (with separate channels for the flue gases and the air) was applied.

About the year 1750, steam heating was invented in England (0.1/0.2 MPa) and toward the end of the nineteenth century in the United States a cast iron boiler and cast iron radiators were introduced. During the first half of the eighteenth century both in England and in France for the first time a heating system was applied with gravitational warm water cycling. Heating with hot water under high pressure was first used in England at the beginning of the eighteenth century, mostly in industry. At the beginning of the twentieth century central heating developed quickly, thanks to the application of pumps (pump heating) [5].

The first municipal heating plant was commissioned in 1877 at Lockport in the United States [5]. In 1887, in the complex buildings of the Technical University of Berlin remote control heating was applied. This system was also applied for the first time in Poland at the beginning of twentieth century, in the buildings of the Technical University of Warsaw [5]. In the year 1900, in Dresden, an extensive heat distribution network was constructed. The first district heating system fed by CHP was commissioned in 1925 in Berlin [5].

Heating systems may be classified as follows [7]:

- with respect to the situation of the source of heat—local, central, and remote central heating,
- with respect to the kind of input energy—fired with coal, gas, fuel oil, municipal wastes, as well as electrical, solar, geothermal heating, and also heating using heat pumps,
- with respect to the kind of heat carriers—heating by means of warm water, hot water, steam, and hot air,
- with respect to transferring the heat–convection, radiation, ventilation, and combined heating.

For local heating the furnace is placed in the room to be heated, and in central heating a single installation (boiler) meets the requirements of all rooms in the whole building. In the case of remote central heating, a heating plant or CHP supplies complex buildings, a district or a whole town with heat. A CHP realizes the cogeneration process of heat and electricity production. There are also special heating systems in which heat pumps, solar energy, geothermal energy, and biogas are utilized. Besides these, so-called territorial systems of industrial waste energy recovery are also applied, in which the recovery installations (waste-heat water boilers and installations of evaporative cooling) cooperate with the district heating systems [11]. In many towns, municipal wastes are utilized, combusted in special boilers installed in the heating plants and CHP plants. Due to the low LHV (3.3/5 MJ/kg) and its widely changing range, the combustion of rubbish is rather difficult.

5.3.2 Consumers of Heat. Heat Carriers

Generally, the consumers of heat are classified as follows:

- municipal consumers,
- complex buildings,
- industry.

Municipal consumers use heat for the purposes of heating, ventilation, air-conditioning, heating up hot tap water, and cooling. The demand for heat for the purposes of heating, ventilation, and air-conditioning varies depending on atmospheric conditions. The principal quantity conditioning the demand for heat for these purposes is the ambient temperature presented in the form of duration curve used to calculate the demand for heat. The duration curves are set up for the respective climatic zones of the country (e.g., Poland has been divided into five climatic zones). The demand for heat required for heating is established based on calculations of the heat losses according to standards concerning various kinds of rooms and various climatic zones. For the purposes of air-conditioning and

ventilation the demand for heat is determined according to the required multiplication factor of the exchange of air in the room.

Complex buildings are concentrated consumers of heat, similar in their character to individual municipal consumers. They are often provided with heat by the same district heating networks. Frequently, however, they are situated at a distance which makes it unprofitable to supply them from remote centrals heating systems, and they are therefore called an “energy islands” in which the demand for energy carriers (including heat) is covered by installations of decentralized energy systems. Heat supplied to complex buildings from district heating networks or their own installations (heating boilers or mini-CHP units) covers the requirements of heating, ventilation, air-conditioning, and production of hot tap water and cooling agents. The character of the heating load depends on the function of the given complex buildings, but to a large extent it is similar to that of individual municipal consumers.

Industrial consumers require heat mainly for technological purposes, its character depending on the specifics of the given technological process. The consumption of heat for ventilation in industrial plants usually exceeds the demand for heat required for heating.

As heat carriers, hot water and steam are usually used. If heat is needed only for the purposes of heating, ventilation, and the production of hot tap water, only hot water is applied. The maximum temperature of water in internal installations amounts to 95 °C because a higher temperature is not admissible due to the fact that it would deteriorate the internal comfort (caused by the radiation of heaters and dry distillation of dust). As a heat carrier, steam is generally used for the purpose of supplying industrial consumers and also some complex buildings (e.g., hospitals).

5.3.3 Heat Demand for Space Heating

In the steady state, the heat demand for heating of the rooms is equal to heat losses through the external walls:

$$\dot{Q}_{\mathrm{h}} = \sum_i A_i k_i (T_{\mathrm{in}} - T_{\mathrm{ex}}) \tag{5.1}$$

where

$\dot{Q}_{\mathrm{h}}$ heat flux for heating,
A_i the surface of external walls of building,
k_i coefficient of heat transfer,
T_{in} internal temperature,
T_{ex} external temperature.

The temperature inside the rooms is determined in compliance with the respective building standards, usually amounting to 20 °C in the case of living

accommodations. The external temperature depends on atmospheric conditions, mostly presented in the form of a duration curve.

The heat demand for heating reaches its maximum when the temperature outside is at its lowest $T_{\text{ex min}}$ characteristic for the given climatic zone:

$$\dot{Q}_{\text{h max}} = \sum_i A_i k_i (T_{\text{in}} - T_{\text{ex min}}) \tag{5.2}$$

Dividing both sides of Eq. (5.1) by Eq. (5.2) and assuming that $k_i = \text{idem}$, we get the reduced relation describing relative heat flux representing the characteristics of the demand for heat:

$$\frac{\dot{Q}_{\text{h}}}{\dot{Q}_{\text{h max}}} = \frac{T_{\text{in}} - T_{\text{ex}}}{T_{\text{in}} - T_{\text{ex min}}} \tag{5.3}$$

Figure 5.1 presents the characteristics of the heat demand for heating purposes.

The maximum heat demand for heating purposes is generally calculated based on the knowledge of the volume of the rooms to be heated and the unit losses of heat determined at ΔTcharacterizing the maximum difference between the inside and minimum outside temperature ($T_{\text{ex min}}$).

$$\dot{Q}_{\text{h max}} = V\dot{q} \tag{5.4}$$

where

V volume of heated rooms,

$\dot{q}$ unit losses of heat in extreme conditions.

The coefficient $\dot{q}$ also takes into account the heat losses due to the infiltration of air (natural ventilation).

5.3.4 Heat Demand for Ventilation

In housing estates the heat demand for natural ventilation is usually taken into account in calculations of the heat required for space heating. In the case of complex buildings and industrial buildings, the heat demand for ventilation depends mainly on the category of consumers and often exceeds the heat demand for heating purposes.

A change of polluted air in complex buildings and industrial buildings is achieved by means of fans which suck in fresh air from outside, preheat it in heaters, and press it into the room in an amount required for the purpose of changing the polluted air. This is called mechanical ventilation. The heat flux required for mechanical ventilation is calculated by means of the equation:

$$\dot{Q}_v = mn_v \left(Mc_p\right)\Big|_{T_{\text{ex}}}^{T_{\text{in}}} (T_{\text{in}} - T_{\text{ex}}) \tag{5.5}$$

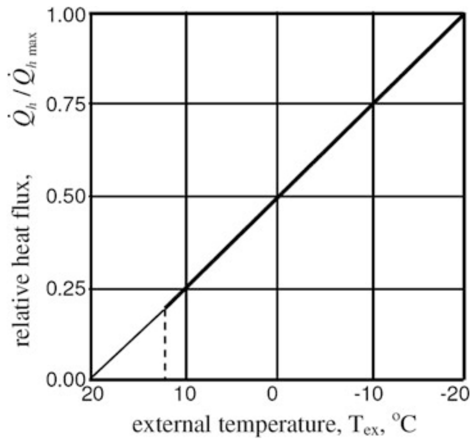


Fig. 5.1 Heat flux for heating

where

$$n_v = \frac{p_{in} V_{in}}{(MR) T_{in}} \tag{5.6}$$

where

p_{in}, T_{in}	pressure and temperature inside ventilated room,
V_{in}	volume (cubature) of ventilated rooms,
(MR)	universal gas constant,
m	multiplicity of the exchange of air per unit of time,
$(Mc_p)\|_{T_{ex}}^{T_{in}}$	mean specific molar heat capacity at constant pressure within the range of the external and internal temperature.

The multiplicity m of the exchange of air is settled by special sanitary regulations. Applying Eq. (5.5) in the case of minimum external temperature, obligatory for ventilation and dividing it on both sides by Eq. (5.5) we get the characteristics of the heat demand for ventilation:

$$\frac{\dot{Q}_v}{\dot{Q}_{v\,max}} = \frac{T_{in} - T_{ex}}{T_{in} - T_{ex\ min\, v}} \tag{5.7}$$

In closed spaces in which production involves large amounts of noxious substances (dust and gases) the minimum external temperature $t_{ex\ min}$ is assumed to be the same as for heating. In other cases, particularly in public complex buildings, where hygienic conditions permit, the minimum external temperature is assumed to amount to −5 °C.

Figure 5.2 presents a diagram of the characteristics of heat demand for ventilation if the minimum external temperature is −5 °C.

In the temperature range −20 °C to −5 °C a smaller amount of air is heated. For this purpose the following methods of regulation are applied:

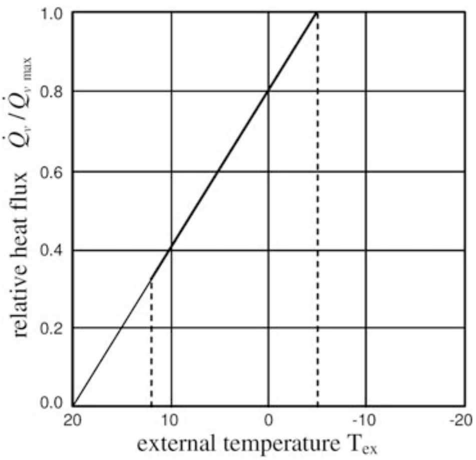


Fig. 5.2 Heat flux for ventilation

- setting ajar the throttles at the sucking orifices,
- reducing the number of revolutions of the fans,
- switching off some parallelly operating fans,
- recirculation of warm air.

If $T_{\text{ex min } v} = T_{\text{ex min}} = -20\ °\text{C}$ the characteristic of the heat demand for ventilation purposes is the same as the characteristic of the heat demand for heating (Fig. 5.1).

5.3.5 Heat Demand for the Production of Hot Tap Water

Calculations concerning the heat demand for the production of hot tap water are based on its unit consumption quoted with respect to the standards obligatory for complex buildings, housing estates, and industrial plants. The demand for hot tap water per day is calculated based on the relation:

$$\bar{G}_{\text{htw}} = nq_m \tag{5.8}$$

where

$\bar{G}_{\text{htw}}$ average daily demand for hot tap water,
n number of consumers,
q_m average calculated daily demand for hot tap water per person.

The flux of heat required to heat up tap water is calculated by means of the relation:

$$\dot{Q}_{\text{htw}} = \dot{G}_{\text{htw}} c_{\text{w}} (T_{\text{htw}} - T_{\text{w}}) \tag{5.9}$$

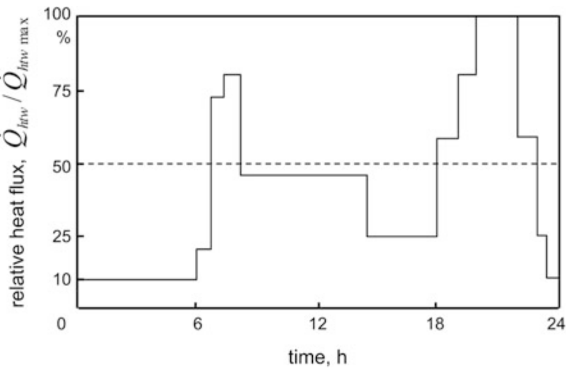


Fig. 5.3 Examplary demand for hot tap water in the case of municipal consumers adapted from Kamler [5]

where

$\dot{G}_{\mathrm{htw}}$ flux of hot tap water,
c_{w} specific heat capacity of water,
T_{htw} temperature of hot tap water,
T_{tw} temperature of tap water.

The temperature of hot tap water at the inlet to the buildings should not exceed 55 °C due to the corrosion of zinc. If the installation consists of plastic, an increase in temperature of the hot tap water above 80 °C accelerates the aging of the material. In Central Europe, the mean temperature of tap water in winter amounts to 7–8 °C and in summer to about 20 °C [5].

Figure 5.3 provides an example of a 24 h diagram illustrating the demand for hot tap water supplied to municipal consumers [5]. In order to minimize fluctuation in the consumption of hot tap water, hot water storage tanks are applied.

5.3.6 Total Demand for Heat

Figure 5.4 presents, for the sake of an example, the way to construct a total duration curve of the heat demand for heating, ventilation, and getting hot tap water [5]. As input data the duration curve of external temperature is assumed to take into account the climatic zone under consideration, as well as the maximum heat demand for heating and ventilation, because both mentioned fluxes of heat depend on the external temperature. The heat demand for heating and ventilation occurs in the heating season, whereas the heat demand for getting hot tap water is assumed to take the form of a double-step diagram presenting the demand for heat, respectively, during and beyond the heating season.

In order to find the total duration curve of the heat demand for space heating, ventilation, and technological purposes (e.g., in complex buildings such as hospitals and rehabilitation centers) the method of composing duration curves must be applied [12], because the heat demand for technological purposes is independent

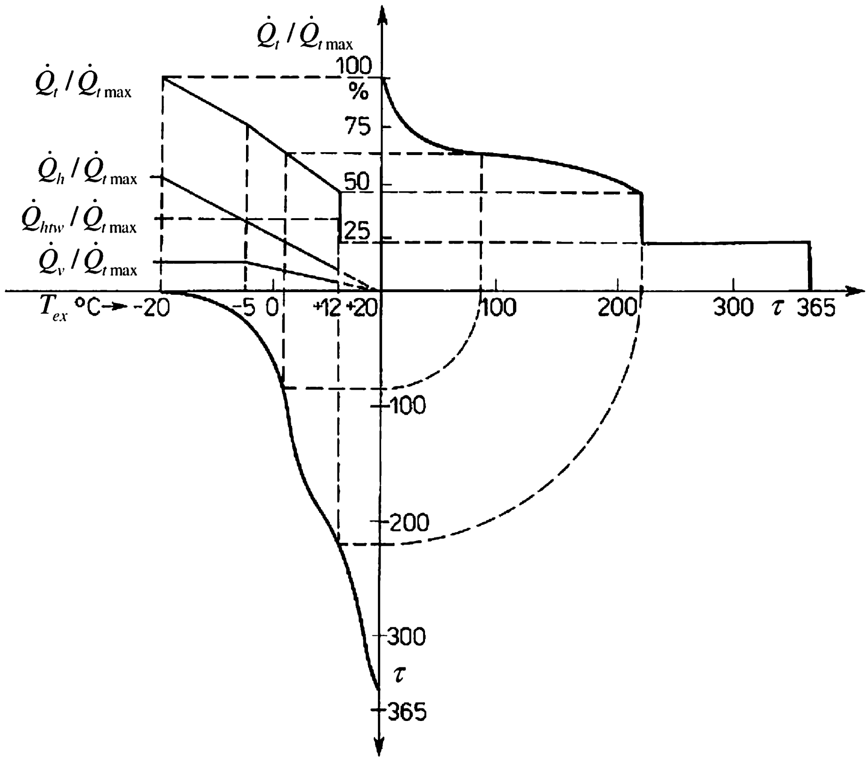


Fig. 5.4 Example of constructing the duration curve of the global heat demand for heating, ventilation, and hot tap water adapted from Kamler [5]; $\dot{Q}_h$-flux of heat required for space heating, $\dot{Q}_v$-flux of heat for ventilation, $\dot{Q}_{htw}$-flux of heat for getting hot tap water, $\dot{Q}_t$-total flux of heat

of the external temperature. Thus, the duration curves of the heat demand for heating and ventilation, on the one hand and that for technological purposes, on the other, ought to be treated as independent variables, applying the method of convolution of the distribution functions of random variables [12].

5.3.7 Choice of Parameters of Heat Carriers

As has already been mentioned, the following heat carriers are applied:

- hot water,
- steam with various parameters (e.g., in hospitals).

If heat is required merely for the purposes of heating, ventilation, and hot tap water, only hot water is used. Hot water is the most appropriate agent for transporting heat at larger distances (up to a score of kilometers). The maximum

parameters of water with respect to its supply and return must be determined by means of economic analysis, which again results from the following thermal analysis. The flux of heat $\dot{Q}_{\mathrm{h}}$ may be expressed by means of the following relations:

(a) with respect to its supply:

$$\dot{Q}_{\mathrm{h}} = \dot{G}_{\mathrm{w}} c_{\mathrm{w}} (T_{\mathrm{h}} - T_{\mathrm{r}}) \tag{5.10}$$

where

$\dot{G}_{\mathrm{w}}$ flux of network water,
c_{w} specific heat capacity of water,
T_{h} temperature of hot water,
T_{r} temperature of return water.

(b) with respect to consumers (direct connection):

$$\dot{Q}_{\mathrm{h}} = A_{\mathrm{h}} k_{\mathrm{h}} \left(\frac{T_{\mathrm{h}} + T_{\mathrm{r}}}{2} - T_{\mathrm{in}} \right) \tag{5.11}$$

where

A_{h} surface of heat transfer,
k_{h} coefficient of heat transfer concerning heaters,
T_{in} internal temperature.

Assuming that the temperature of hot water is constant (e.g., restricted by the pressure of heating steam in the heat exchanger) and that $\dot{Q}_{\mathrm{h}} = \text{idem}$, the drop in the temperature of the return water is accompanied by a possible decrease of the water flux $\dot{G}_{\mathrm{w}}$ Eq.(5.10) and thus also by a decreased diameter of the pipelines of district heating networks and decreased capital expenditures. On the other hand, however, the mean temperature of the heating agent also decreases so that the surface of the heaters must be increased (higher capital expenditure) in order to transfer the same flux of heat. Both of these contrary factors determine the choice of the optimal values of the maximum temperature of hot water and return water.

Additional factors which ought to be taken into account in the choice of the optimal values of the temperature of hot water and return water are:

- power rating of the pumps and the consumption of electricity for the purpose of pumping the network water; the smaller the flux of pumping, the smaller are the power rating of the pumps and the consumption of electricity,
- losses occurring during the transport of heat; the higher the temperature of hot water and the return water the higher are the losses of heat,
- the cogeneration factor when heat is supplied from CHP plant; the lower temperature of hot water corresponds to a higher cogeneration factor.

Taking into account the aforementioned factors resulting from thermal analysis the maximum values of the temperature of hot and return water are determined by means of economic optimization [10].

5.3.8 Thermal Centers

The heat agent is supplied to the consumers (buildings and technical installations) by heating networks. Then, it must be distributed to the respective installations (heating and ventilation systems and technological heat exchangers). The installations are frequently adapted to other parameters than those of the external networks. The high thermal parameters of the external networks are changed to lower parameters of the internal networks by means of thermal centers. A thermal center comprises a set of segments of pipelines and thermal devices with pipe fittings and measuring instruments, from the cutoff valve of the external network to the valve cutting off the thermal center from the internal installations.

The aims of the thermal center are to:

- transfer heat from the external network to internal installations,
- initiate the circulation of the heating agent in the internal network,
- protect internal installations against increases in pressure exceeding the admissible value,
- reduce the temperature and pressure of the heating agent,
- retain the contaminations.

The following ways to connect the consumers of heat with district heating networks can be distinguished:

- direct low-parameter connection, when the thermal parameters of the heating agent supplied by the external network need not be reduced,
- direct connection by means of a jet pump, in which the parameters (temperature and pressure) of the water from the external networks are reduced,
- direct connection by means of pumps,
- indirect connection by means of heat exchangers, mainly of the type water/water, and rarely steam/water.

Direct thermal centers are nowadays rather rarely used and are displaced by indirect thermal centers, due to their drawbacks such as infiltration of contaminations into the internal installations, the necessity of maintaining an adequate disposable pressure in the thermal center, and difficulties in warranting a stable hydraulic control.

Indirect thermal centers (Fig. 5.5) are usually installed as two- or three-functional ones (heating and ventilation, as well as hot tap water) [5]. Mostly countercurrent heat exchangers are applied, either tubular or lamellar.

The standard equipment of the thermal centers comprises:

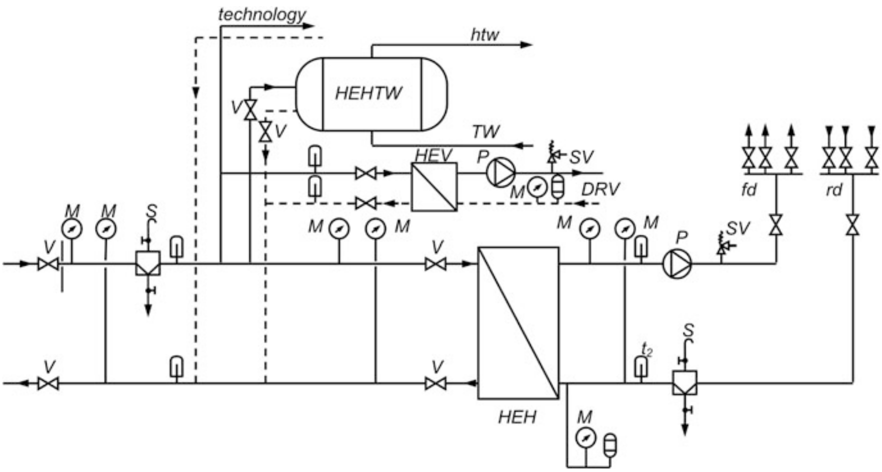


Fig. 5.5 Scheme of a multi-functional thermal center. Denotations: *V* cutoff valve, *M* manometer, *S* sludger, *SV* safety-valve, *DRV* diaphragmatic rising vessel, *T* thermometer, *fd* feeding divider, *rd* returned divider, *P* circulating pump, *HEHTW* heat exchanger for hot tap water, *htw* hot tap water, *TW* tap water, *HEV* heat exchanger for ventilation, *HEH* heat exchanger for central heating

- sludger of network water,
- thermometers and manometers,
- heat exchangers,
- circulating pumps of heating water and hot tap water,
- controlling orifices,
- elements of automatics,
- heat meters.

In the thermal centers of hot tap water accumulating preheaters operate, included in the heating network either in series or in parallel.

5.3.9 Control of the Supply of Heat

The heat demand for heating and ventilation depends on the external temperature, and the heat demand for technological purposes (e.g., in hospitals) depends on the operating conditions of the technological devices. Therefore, the supply of heat must be properly controlled.

Depending on the place where the control is realized we can distinguish:

- central control (concerning production),
- local control (in receivers or thermal centers).

Central control is applied when the thermal load of the heating network is homogeneous (e.g., merely heating). Local control is applied when the consumption varies (heating, ventilation, and hot tap water).

The amount of heat Q transferred by thermal devices is expressed by the relation:

$$Q = Ak\Delta T\tau \tag{5.12}$$

where

A surface of heat transfer,
k coefficient of heat transfer,
ΔT mean difference of temperature,
τ time.

Analyzing the ways of controlling the mean arithmetic difference in the temperature may be applied:

$$\Delta T = \frac{T_h + T_r}{2} - \frac{T_1 + T_2}{2} \tag{5.13}$$

where

T_h temperature of hot water in external heating network,
T_r temperature of returned water of external heating network,
T_1 inlet temperature of water in internal heating network,
T_2 outlet temperature of water in internal heating network.

Including Eqs. (5.10) and (5.13) into Eq. (5.12) we get:

$$Q = \frac{T_h - \frac{1}{2}(T_1 + T_2)}{\frac{1}{Ak} + \frac{1}{2\dot{G}c_w}}\tau \tag{5.14}$$

Analyzing the relation (5.14) five possibilities of control can be distinguished:

- local control by changing the surface of the receivers (partial switching off); rarely used, usually in technological installations,
- changing the heat transfer coefficient by changing the flow rate of the heating agent or the application of shields (local control),
- changing the temperature of hot water in the external heating network (qualitative central control),
- changing the flux of hot water in the external heating network (quantitative central control),
- changing the time of operation of the installations, introducing breaks in the operation (so-called pulsatory control); this kind of control is applied in interim periods in the case of a higher external temperature and the necessity of maintaining an adequate temperature of the feeding hot water in order to obtain hot tap water with a constant temperature.

In most cases qualitative control in the water heating network is applied. The advantages of such a control are:

- easy attendance and exploitation,
- the possibility of maintaining a lower temperature of the feeding water for a longer time (higher effects thanks to cogeneration due to the lower pressure of the heating steam).

A drawback of the qualitative control is the fact that it does not strictly meet the needs of all the consumers. Practically, in this system the temperature is adjusted to the needs of the dominant number of consumers. For the remaining consumers additional local control is applied.

The flux of heat transferred by the external heating network is expressed by the Eq. (5.10), based on which the methods of central control are defined, viz.:

- qualitative control-$\dot{G}_{\mathrm{w}} = \mathrm{idem},\ T_{\mathrm{h}}, T_{\mathrm{r}} =\ \mathrm{varia}$,
- quantitative control- $\dot{G}_{\mathrm{w}} =\ \mathrm{varia},\ T_{\mathrm{h}} = \mathrm{idem},\ T_{\mathrm{r}} =\ \mathrm{varia}$,
- hybrid quantitative-qualitative control.

Qualitative control, realized by maintaining a constant flow rate ($\dot{G}_{\mathrm{w}} = \mathrm{idem}$), is characterized by keeping the hydraulic conditions of the heating networks stable. The third way of central control is applied in heating networks which supply heat and hot tap water in the heating season and only hot tap water in summer.

5.3.9.1 Qualitative Control

Characteristics of the heating network in the case of supplying heat

Based on Eqs. (5.1), (5.10), and (5.11) concerning running conditions and minimum external temperature (maximum heating load), dividing both sides of them by themselves and assuming that the heat transfer coefficients k and k_{h} are constant, in the case of direct connection without mixing, we get:

$$\frac{\dot{Q}_{\mathrm{h}}}{\dot{Q}_{\mathrm{h\,max}}} = \frac{T_{\mathrm{in}} - T_{\mathrm{ex}}}{T_{\mathrm{in}} - T_{\mathrm{ex\;min}}} = \frac{T_{\mathrm{h}} - T_{\mathrm{r}}}{T_{\mathrm{h\;max}} - T_{\mathrm{r\;max}}} = \frac{\frac{1}{2}(T_{\mathrm{h}} + T_{\mathrm{r}}) - T_{\mathrm{in}}}{\frac{1}{2}(T_{\mathrm{h\;max}} + T_{\mathrm{r\,max}}) - T_{\mathrm{in}}} \tag{2}$$

hence

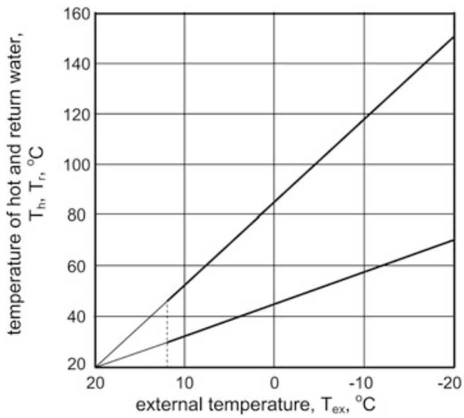
$$T_{\mathrm{h}} = T_{\mathrm{in}} + \frac{T_{\mathrm{h\,max}} - T_{\mathrm{in}}}{T_{\mathrm{in}} - T_{\mathrm{ex\,min}}}(T_{\mathrm{in}} - T_{\mathrm{ex}}) \tag{5.16}$$

$$T_{\mathrm{r}} = T_{\mathrm{in}} + \frac{T_{\mathrm{r\,max}} - T_{\mathrm{in}}}{T_{\mathrm{in}} - T_{\mathrm{ex\,min}}}(T_{\mathrm{in}} - T_{\mathrm{ex}}) \tag{5.17}$$

where

T_{in} internal temperature,
T_{ex} running external temperature,
$T_{\mathrm{ex\;min}}$ minimum external temperature,

Fig. 5.6 Characteristics of heating network (qualitative control)



$T_{h\,max}$ maximum temperature of hot water,
$T_{r\,max}$ maximum temperature of returned water.

Figure 5.6 presents the characteristics of the heating network concerning heating loads according to Eqs. (5.16) and (5.17), valid in the case of direct connections without mixing and assuming that the heat transfer coefficient for the heaters is constant.

In the case of indirect connections by means of a heat exchanger of the type water/water the energy balance equation is additionally applied and the relation determining the power rating of the heat exchanger is calculated in compliance with the method of thermal effectivity [9].

$$\dot{Q} = \dot{G}_w c_w (T_h - T_r) = \dot{G}_{hr} c_{hr} (T_1 - T_2) \tag{5.18}$$

$$\dot{Q} = \varepsilon \dot{W}_{min} (T_h - T_2) \tag{5.19}$$

where

$\dot{G}_{hr}$ flux of water in installations of heat receivers,
c_{hr} specific heat capacity of water in installation of heat receivers,
ε effectivity of heat exchanger,
W_{min} smaller value of flux of heat capacity concerning water from external heating network and water in installation of heat receivers.

As far as the internal heating network is concerned, feeding the receiving installations, Eqs. (5.16) and (5.17) are valid for the qualitative control, whereby T_h and T_r are to be replaced, respectively, by T_1 and T_2. Then in the case of the external heating network based on the Eqs. (5.18) and (5.19) and the characteristics of the internal heating network, the following equations concerning the temperature T_h, T_r of heat and return water are obtained:

$$T_{\mathrm{h}} = T_{\mathrm{in}} + \left[T_{2\,\mathrm{max}} - T_{\mathrm{in}} + \frac{\dot{G}_{\mathrm{hr}} c_{\mathrm{hr}}}{\varepsilon \dot{W}_{\min}} (T_{1\,\mathrm{max}} - T_{2\,\mathrm{max}})\right] \frac{T_{\mathrm{in}} - T_{\mathrm{ex}}}{T_{\mathrm{in}} - T_{\mathrm{ex\,min}}} \tag{5.20}$$

$$T_{\mathrm{r}} = T_{\mathrm{in}} + \left[T_{2\,\mathrm{max}} - T_{\mathrm{in}} + \left(\frac{\dot{G}_{\mathrm{w}} c_{\mathrm{w}}}{\varepsilon \dot{W}_{\min}} - 1\right)(T_{\mathrm{h\,max}} - T_{\mathrm{r\,max}})\right] \frac{T_{\mathrm{in}} - T_{\mathrm{ex}}}{T_{\mathrm{in}} - T_{\mathrm{ex\,min}}} \tag{5.21}$$

These two equations are valid assuming that the heat transfer coefficients of the heaters are constant.

Characteristics of the heating network in the case of ventilation

Based on Eqs. (5.7) and (5.10) and the relations of the amount of heat transferred from the air heater:

$$\dot{Q}_v = \sum A_{\mathrm{h}} k_{\mathrm{h}} \left(\frac{T_{\mathrm{h}} + T_{\mathrm{r}}}{2} - \frac{T_{\mathrm{in}} + T_{\mathrm{ex}}}{2}\right) \tag{5.22}$$

expressing the running and extreme conditions and assuming that the heat transfer coefficients are constant, we get:

$$\frac{\dot{Q}_v}{\dot{Q}_{v\,\max}} = \frac{T_{\mathrm{in}} - T_{\mathrm{ex}}}{T_{\mathrm{in}} - T_{\mathrm{ex\,min}\,v}} = \frac{T_{\mathrm{h}} - T_{\mathrm{r}}}{T_{\mathrm{h\,max}} - T_{\mathrm{r\,max}}} = \frac{T_{\mathrm{h}} + T_{\mathrm{r}} - T_{\mathrm{in}} - T_{\mathrm{ex}}}{T_{\mathrm{h\,max}} + T_{\mathrm{r\,max}} - T_{\mathrm{in}} - T_{\mathrm{ex\,min}\,v}} \tag{5.23}$$

hence in the range of varying heat consumption:

$$T_{\mathrm{h}} = T_{\mathrm{in}} + \frac{T_{\mathrm{h\,max}} - T_{\mathrm{in}}}{T_{\mathrm{in}} - T_{\mathrm{ex\,min}\,v}} (T_{\mathrm{in}} - T_{\mathrm{ex}}) \tag{5.24}$$

$$T_{\mathrm{r}} = T_{\mathrm{in}} + \frac{T_{\mathrm{r\,max}} - T_{\mathrm{in}}}{T_{\mathrm{in}} - T_{\mathrm{ex\,min}\,v}} (T_{\mathrm{in}} - T_{\mathrm{ex}}) \tag{5.25}$$

When $T_{\mathrm{ex}} < T_{\mathrm{ex\,min}\,v}$ the relations hold true:

$$\frac{\dot{Q}_v}{\dot{Q}_{v\,\max}} = 1 \tag{5.26}$$

hence taking into account Eq. (5.23) we get:

$$T_{\mathrm{h}} = T_{\mathrm{h\,max}} \tag{5.27}$$

$$T_{\mathrm{r}} = T_{\mathrm{r\,max}} \tag{5.28}$$

Figure 5.7 provides exemplary characteristics of the heating network in the case of ventilation.

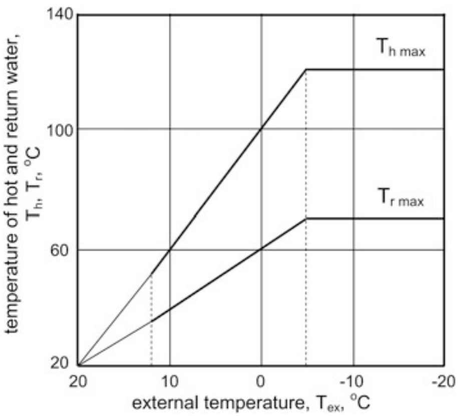


Fig. 5.7 Characteristics of the heating network for ventilation (qualitative control)

5.3.9.2 Quantitative Control

Characteristics of heat loads

The water flux is controlled by throttling its flow. The result of changing the flux of water is a change in the temperature of the return water. The set of equations that determine the characteristics of the heating network, based on the assumption that the coefficients of heat transfer are constant and that the consumers are connected directly without mixing, takes the form:

$$\frac{\dot{Q}_{\mathrm{h}}}{\dot{Q}_{\mathrm{h\ max}}}=\frac{T_{\mathrm{in}}-T_{\mathrm{ex}}}{T_{\mathrm{in}}-T_{\mathrm{ex\ min}}}=\frac{\dot{G}_{\mathrm{w}}(T_{\mathrm{h\ max}}-T_{\mathrm{r}})}{\dot{G}_{\mathrm{w\ max}}(T_{\mathrm{h\ max}}-T_{\mathrm{r\ max}})}=\frac{\frac{1}{2}(T_{\mathrm{h\ max}}+T_{\mathrm{r}})-T_{\mathrm{in}}}{\frac{1}{2}(T_{\mathrm{h\ max}}+T_{\mathrm{r\ max}})-T_{\mathrm{in}}} \tag{5.29}$$

where

$\dot{G}_{\mathrm{w}}$ running flux of water in the heating network,

$\dot{G}_{\mathrm{w\ max}}$ maximum flux of water in the heating network.

From Eq. (5.29) we get:

$$T_{\mathrm{r}}=2T_{\mathrm{in}}-T_{\mathrm{h\ max}}+(T_{\mathrm{h\ max}}+T_{\mathrm{r\ max}}-2T_{\mathrm{in}})\frac{T_{\mathrm{in}}-T_{\mathrm{ex}}}{T_{\mathrm{in}}-T_{\mathrm{ex\ min}}} \tag{5.30}$$

$$\frac{\dot{G}_{\mathrm{w}}}{\dot{G}_{\mathrm{w\ max}}}=\frac{T_{\mathrm{in}}-T_{\mathrm{ex}}}{T_{\mathrm{in}}-T_{\mathrm{ex\ min}}}\cdot\frac{T_{\mathrm{h\ max}}-T_{\mathrm{r\ max}}}{T_{\mathrm{h\ max}}-T_{\mathrm{r}}} \tag{5.31}$$

Characteristics of ventilation loads

Assuming that the coefficient of heat transfer for heaters is constant and when the consumers are connected directly, in the case of a varying range of heat consumption we get:

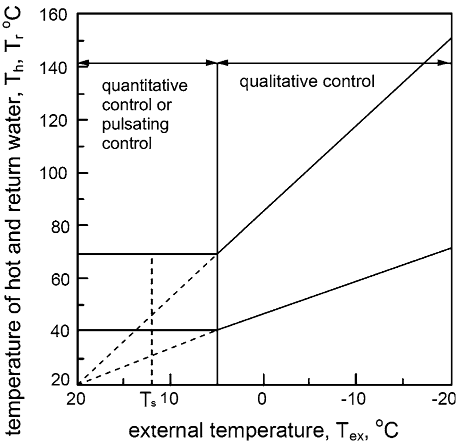


Fig. 5.8 Characteristics of heating network in the case of hybrid control; T_s temperature of the start of heating season

$$\frac{\dot{Q}_v}{\dot{Q}_{v\,\max}} = \frac{T_{\rm in} - T_{\rm ex}}{T_{\rm in} - T_{{\rm ex}\,\min\,v}} = \frac{\dot{G}_{\rm w}}{\dot{G}_{{\rm w}\,\max}} \frac{(T_{\rm h\,max} - T_{\rm r})}{(T_{\rm h\,max} - T_{\rm r\,max})}$$
$$= \frac{(T_{\rm h\,max} + T_{\rm r}) - (T_{\rm in} + T_{\rm ex})}{(T_{\rm h\,max} + T_{\rm r\,max}) - (T_{\rm in} + T_{ex\,\min\,v})} \tag{5.32}$$

and hence:

$$T_{\rm r} = (T_{\rm in} + T_{\rm ex}) - T_{\rm h\,max} + \frac{\dot{Q}_v}{\dot{Q}_{v\,\max}}[(T_{\rm h\,max} + T_{\rm r\,max}) - (T_{\rm in} + T_{{\rm ex}\,\min\,v})] \tag{5.33}$$

In the considered case a formula analogical to (5.31) is valid, in which only $T_{\rm ex\,min}$ is replaced by $T_{{\rm ex}\,\min\,v}$.

5.3.9.3 Hybrid Control

Qualitative control is usually restricted to one kind of consumers (mainly for heating purposes). If there are various consumers (e.g., of heating and hot tap water) such a control is insufficient. When hot water is used for the production of hot tap water, its temperature must amount to at least 60/70 °C. Thus, in the characteristics of the heating network two ranges of control are to be distinguished (Fig. 5.8): qualitative and quantitative or pulsating control. The application of pulsating control is based on the cumulative ability of the rooms.

A quantitative–qualitative control of the heating network can also be realized by distinguishing several ranges of external temperature in the characteristics of the heating network [6]. An increase in the external temperature to the successive ranges corresponds to a decreasing flux of network water changing stepwise from range to range. Within each range qualitative control is being realized. This method of control is advantageous because the consumption of electrical energy

for driving the pumps is reduced, but on the other hand it does not warrant a stability of hydraulic conditions in the heating network.

5.3.10 Application of Heat Pumps in Heat Engineering

Most often compression heat pumps driven by electrical energy are used. They can draw heat from the atmosphere, water reservoirs, the ground, warm sewage, or solar collectors. Making use of the atmosphere is least effective because the heat demand for heating increases with a drop of the air temperature and simultaneously the COP (Coefficient of Performance) of the heat pump decreases. A large water reservoir (e.g., a lake) is a better bottom source of heat, thanks to its stable temperature under the cover of ice. Extracting heat from the ground is feasible when the temperature of the soil is higher than the temperature of the atmosphere. The best bottom sources of heat are resources of low-temperature waste energy (sewage and waste heat) and water preheated in solar collectors. The higher the temperature of the bottom source of heat, the higher the COP.

The condition of energy effectiveness in the application of compression heat pumps takes the form [11]:

$$\mathrm{COP} > \frac{\eta^*_{\mathrm{Eh}}}{\eta^*_{\mathrm{Eel}}} \tag{5.34}$$

where

COP Coefficient of Performance of a heat pump,

η^*_{Eh} cumulative energy efficiency of heat production in a source replaced by a heat pump,

η^*_{Eel} cumulative energy efficiency of electricity production.

In this equation the losses occurring in the course of transporting the heat from the heat pump have been neglected, and the cumulative investment component has not been taken into account.

The fact that energy effectivity has been satisfied does not mean that the application of an electrically driven compression heat pump is profitable, because its installation is connected with considerable expenditures, not only direct, but also additional expenditures, due to an increased power rating of the power station and electrical transmission grid. This is of essential importance in the case of thermal power plants. In the case of hydro-electric power plants it is less important.

The application of an internal combustion engine in the compression heat pump may improve the energy effectiveness because the heat produced by cooling the combustion engine and the physical enthalpy of flue gases are additionally utilized [13]. Figure 5.9 presents the band chart of the energy balance of a compression heat pump driven by an internal combustion engine. The energy effectiveness of a

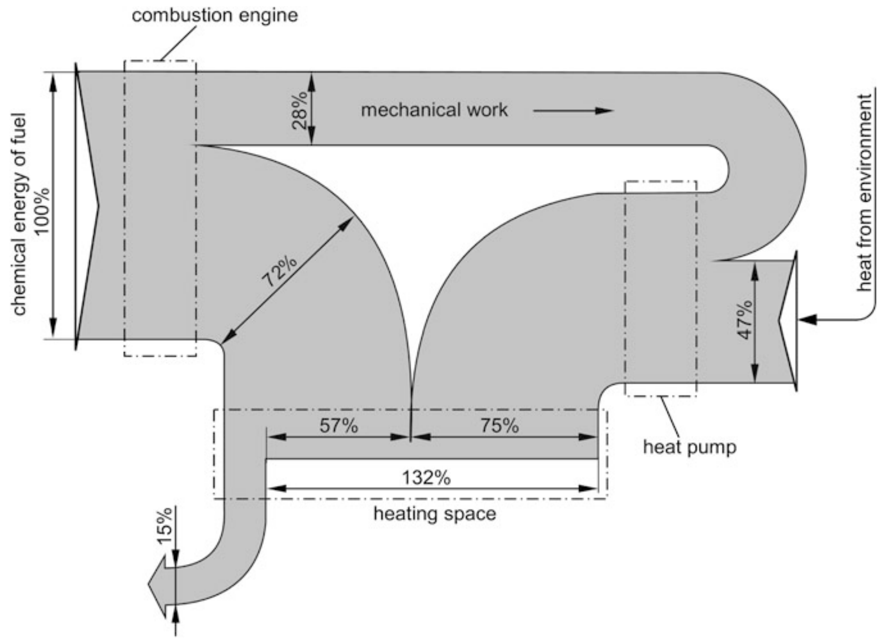


Fig. 5.9 Band chart of the energy balance of a compression heat pump driven by an internal combustion engine [13]

compression heat pump driven by a combustion engine may be expressed by the formula:

$$\varepsilon_{\mathrm{Eie}} = \frac{\dot{Q}_{\mathrm{h}}}{\dot{E}_{\mathrm{ch}}} = \mathrm{COP}\ \eta_{\mathrm{Eie}} + (1 - \eta_{\mathrm{Eie}})\eta_{\mathrm{wh}} \tag{5.35}$$

where

$\dot{Q}_{\mathrm{h}}$ useful heat flux,
$\dot{E}_{\mathrm{ch}}$ chemical energy of fuel,
η_{Eie} energy efficiency of internal combustion engine,
η_{wh} energy efficiency of installation of waste heat recovery.

Assuming, for instance, that $\mathrm{COP} = 2,7$; $\eta_{\mathrm{Eie}} = 0,28$; $\eta_{\mathrm{wh}} = 0,8$ we get $\varepsilon_{\mathrm{Eie}}=$ 1,32. The amount of heat supplied by the aggregate heat pump and internal combustion engine is definitely greater than the used chemical energy of fuel. Besides the expenditures, the economic effectiveness of such a solution is influenced by the higher cost of liquid or gaseous fuels.

Also, the application of an absorption heat pump can lower the threshold of energy effectivity. Ignoring the losses of transferring heat from the heat pump and neglecting the cumulative investment component, the condition of the energy effectivity of applying absorption heat pumps takes the form:

$$\mathrm{COP} > \frac{\eta^*_{\mathrm{Eh}}}{\eta^*_{\mathrm{E\,hd}}} \tag{5.36}$$

where $\eta^*_{\mathrm{E\,hd}}$ denotes cumulative energy efficiency of the production of heat driving the absorption heat pump.

The condition (5.36) is particularly easily satisfied when the replaced installation is a boiler house, which can simultaneously be the source of heating steam for the absorption heat pump. In this case the right-hand side of the inequality (5.36) approaches 1, and the COP for the absorption heat pump is evaluated as $\sim 1.4/1.7$. When the replaced installation is a CHP plant, the condition (5.36) is more difficult to be satisfied.

The energy effectivity of applying heat pumps can be improved by combining the heat pump with a refrigerator (cooling-heating cogenerating system), whereby heat is rather a by-product [11].

5.4 Cogeneration of Heat and Electricity: Combined Heat and Power (CHP)

5.4.1 Thermodynamic Motivation of Benefits Resulting from the Realization of Heat-and-Power Cogeneration

The irreversibility of thermal processes can be decreased by combining them for the purpose of reducing the number of thermodynamic processes [10]. Simultaneously, the emission of noxious substances to the environment can be reduced. Electricity and heat may be generated in separate processes, i.e., in the power station and in the heating plant. In a power station the cycle of the engine is realized. The heating plant may be considered as the cycle of a heat pump. Figure 5.10 presents irreversible cycles of the engine and heat pump [11]. Both of these individual cycles consist of isobaric processes of heat transfer from the working fluid to the environment. As the directions of heat transfer are with respect to the environment opposite to each other, the intermediate role of environment as a source of heat may be rejected. In this way we can eliminate two irreversible processes of heat transfer and consequently also the capital investment for heat exchangers. This can be achieved by shifting the upper isobar of the heat pump to the cycle of the engine, realizing in this way a combined engine and heating cycle [11, 12]. Thus, two irreversible processes of compression and expansion realized in the cycle of the heat pump are also eliminated. Furthermore, the ranges of irreversible compression and expansion in the cycle of the engine are reduced. The combined engine and heating cycle is shown in Fig. 5.10b. This cycle is practically realized in CHP plants. Figure 5.11 presents the diagram of a classical CHP plant equipped with a back-pressure turbine.

The second argument for cogeneration results from the comparison of the energy and exergy efficiencies of the boilers. Figure 5.12 presents a nomogram of

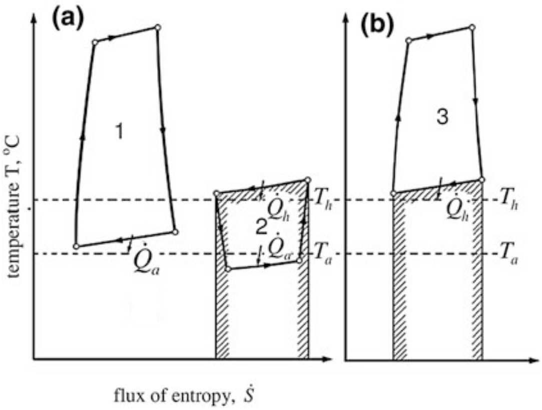


Fig. 5.10 The idea of a CHP plant [11]; $\dot{Q}_h$ flux of heat for heating; $\dot{Q}_a$ flux of heat exchanged between working fluid and environment, T_h temperature of heating space, T_a ambient temperature; *1* cycle of thermal engine; *2* cycle of heat pump; *3* cycle of CHP

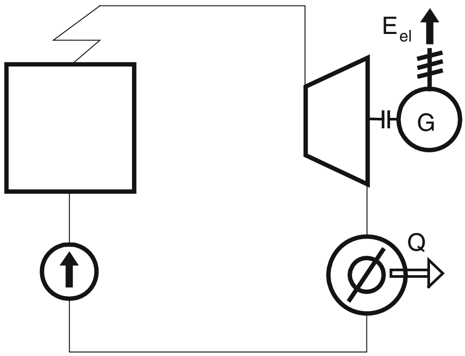


Fig. 5.11 Classical CHP plant equipped with a back-pressure turbine

the interdependence between the energy and exergy efficiency of steam boilers [16]. Boilers are characterized by the highest exergy losses of all elements of the power station. These losses can be reduced by increasing the thermal parameters of live steam. In CHP plants, the pressure and temperature of live steam are much higher than the thermal parameters in boilers of heating plants, particularly those plants equipped with water heater boilers. In this way we can decrease exergy losses, thanks to the realization of combined heat and electricity production.

The third thermodynamic argument for cogeneration is the possibility of a partial compensation of heat losses while transporting the heat. Figure 5.13 presents duration curves of the net and gross heat demand. Assuming that the share of cogeneration amounts to 0.6 we see the area of the cogeneration part and the area of the non-cogeneration part of a CHP (e.g., water heater boiler). In the range of peak loads, the transportation losses of heat are covered by non-cogeneration installations, whereas at lower loads the compensation of the transportation losses of heat is achieved by increasing the production of heat in cogeneration. An increase in the production of heat in the cogeneration process due to heat losses affects the increase of electricity production and the growth of the useful effects of cogeneration. In this way the heat losses can be partially compensated.

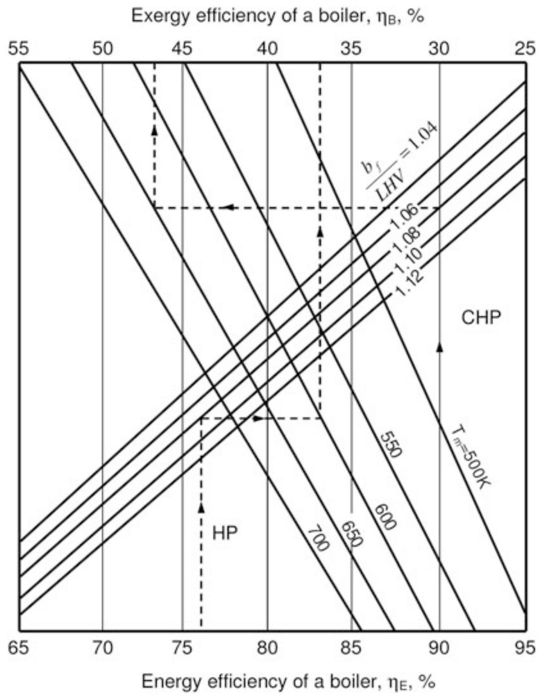


Fig. 5.12 Energy and exergy efficiency of a steam boiler; *CHP* plant, *HP* heating plant, b_f exergy of fuel, T_m thermodynamic average temperature (Appendix C)

5.4.2 Energy Effects of Heat-and-Power Cogeneration in CHP Plants

Savings in the chemical energy of fuel are determined by comparing its consumption in heating plants and power stations operating separately with its consumption in CHP plants. This comparison is carried out based on the assumption that the demands for heat and electricity by the consumers are the same in both cases of production:

$$Q_0 = \text{idem} \qquad \text{and} \qquad E_{\text{el }0} = \text{idem}$$

where

Q_0 demand for heat by consumers,

$E_{\text{el }0}$ demand for electricity by consumers.

The consumption of the chemical energy of fuel in the former case is the sum of its consumption in the heating plant and power station

$$E_{ch\,s} = \frac{Q_0}{\eta_{\text{E hp}}\eta'_{\text{ht}}} + \frac{E_{\text{el }0}}{\eta_{\text{E pp}}\eta'_{\text{et}}} \tag{5.37}$$

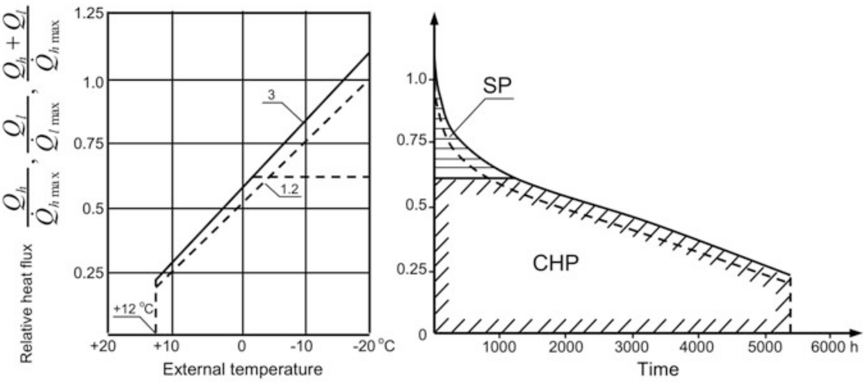


Fig. 5.13 The effect of heat losses on the characteristics of heat load and duration curve of heat demand: *1* relative net heat load, *2* relative losses of heat transport, *3* relative gross heat load, *SP* devices of separate heat production (pressure reducing valve or water heater), *CHP* plant

where

$\eta_{\mathrm{E\,hp}}$ net energy efficiency of heating plant,
$\eta_{\mathrm{E\,pp}}$ net energy efficiency of power plant,
η'_{ht} energy efficiency of heat transmission concerning heating plant,
η'_{et} energy efficiency of electricity transmission concerning power plant.

In the latter case the consumption of the chemical energy of fuel results from equation:

$$E_{\mathrm{ch\,CHP}} = \frac{1}{\eta_{\mathrm{E\,CHP}}}\left[\frac{Q_0}{(1-\varepsilon_{\mathrm{h}})\eta_{\mathrm{ht}}} + \frac{E_{\mathrm{el}\,0}}{(1-\varepsilon_{\mathrm{el}})\eta_{\mathrm{et}}}\right] \tag{5.38}$$

where

$\eta_{\mathrm{E\,CHP}}$ gross energy efficiency of CHP ($\eta_{\mathrm{E\,CHP}} = \mathrm{EUF}$ [3]),
EUF energy utilization factor,
η_{ht} energy efficiency of heat transmission,
η_{et} energy efficiency of electricity transmission,
$\varepsilon_{\mathrm{h}}, \varepsilon_{\mathrm{el}}$ indices of own consumption of heat and electricity.

Savings of the chemical energy of fuels achieved by heat-and-power cogeneration is determined by the following formula:

$$-\Delta E_{\mathrm{ch}} = Q_0\left[\frac{1}{\eta_{\mathrm{E\,hp}}\eta'_{\mathrm{ht}}} - \frac{1}{\eta_{\mathrm{E\,CHP}}(1-\varepsilon_{\mathrm{h}})\eta_{\mathrm{ht}}}\right] + E_{\mathrm{el}\,0}\left[\frac{1}{\eta_{\mathrm{E\,pp}}\eta'_{\mathrm{et}}} - \frac{1}{\eta_{\mathrm{E\,CHP}}(1-\varepsilon_{\mathrm{el}})\eta_{\mathrm{et}}}\right] \tag{5.39}$$

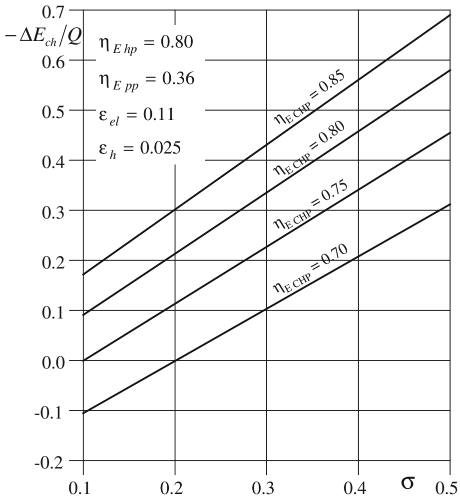


Fig. 5.14 Index of the saving of chemical energy of fuel

Assuming that the efficiency of heat and electricity transmission is the same in both cases and dividing Eq. (5.39) by the gross amount of heat, we get:

$$\frac{-\Delta E_{\mathrm{ch}}}{Q} = \left[\frac{1}{\eta_{\mathrm{E\ hp}}} - \frac{1}{\eta_{\mathrm{E\ CHP}}(1-\varepsilon_{\mathrm{h}})}\right](1-\varepsilon_{\mathrm{h}}) + \sigma\left[\frac{1}{\eta_{\mathrm{E\ pp}}} - \frac{1}{\eta_{\mathrm{E\ CHP}}(1-\varepsilon_{\mathrm{el}})}\right](1-\varepsilon_{\mathrm{el}}) \tag{5.40}$$

in which the index of cogeneration (ratio of electricity to heat produced in cogeneration) is:

$$\sigma = \frac{E_{\mathrm{el}}}{Q} \tag{5.41}$$

where Q and E_{el} denote gross production of heat and electricity in CHP plant.

Figure 5.14 presents the dependence of the saving index of the chemical energy of fuel on the index of cogeneration and gross energy efficiency of the CHP. Figure 5.14 shows, for instance, that if 100 MW of heat is produced in a CHP plant with the efficiency $\eta_{\mathrm{E\ CHP}}$= 0.85 and the coefficient of cogeneration $\sigma = 0.4$, we may expect savings in the chemical energy of fuel to be about 56 MW .

5.4.3 The Share of the Fuel Consumption in the Production of Heat and the Production of Electricity

The division of the consumption of the chemical energy of fuel in CHP between the production of heat and the production of electricity is connected with the determination of the partial efficiencies of heat and electricity production. In order

to determine these partial efficiencies in the cogeneration process the method of avoided expenditure of fuels is applied. This is the same idea as the method of avoided cost being applied to divide the costs [10, 11]. This method is also called *method of a replaced process.* Electricity produced as a by-product replaces partially the electricity from system power stations. Therefore, the production of electricity in CHP plants should be charged with the same rate as the consumption of chemical energy of fuel as in the replaced (reference) power station. The term “replaced” should, from the viewpoint of Directive on the promotion of cogeneration [1], be understood as a reference power station. The partial gross energy efficiency $\eta_{\mathrm{E\ el\ CHP}}$ of the production of electricity in CHP plants are determined based on the relation:

$$\eta_{\mathrm{E\ el\ CHP}} = \frac{N_{\mathrm{el\ CHP}}}{\left(\dot{P}\,\mathrm{LHV}\right)_{\mathrm{el\ CHP}}} \tag{5.42}$$

where

$N_{\mathrm{el\ CHP}}$ electrical power in cogeneration,

$\dot{P}\mathrm{LHV}_{\mathrm{el\ CHP}}$ consumption of the chemical energy of fuel charging electricity production.

Power replaced in a system power plant results from the condition of equality of the electrical power reaching the consumer (loco consumer):

$$\overline{N}_{\mathrm{el\ CHP\ (lo)}} = \overline{N}_{\mathrm{el\ pp\ (lo)}} \tag{5.43}$$

where

$\overline{N}_{\mathrm{el\ CHP\ (lo)}}$ electrical power from CHP (reaching the consumers),

$\overline{N}_{\mathrm{el\ pp\ (lo)}}$ electrical power from power plant (reaching the consumers).

Relation (5.43) may also be expressed as:

$$N_{\mathrm{el\ CHP}}(1 - \varepsilon_{\mathrm{el}})\eta_{\mathrm{et}} = N_{\mathrm{el\ pp}}\eta'_{\mathrm{et}} \tag{5.44}$$

where $N_{\mathrm{el\ pp}}$ denotes the net electrical power of the power plant.

Applying the method of avoided input energy we get the relation:

$$\left(\dot{P}\mathrm{LHV}\right)_{\mathrm{el\ CHP}} = \frac{N_{\mathrm{el\ pp}}}{\eta_{\mathrm{E\ pp}}} \tag{5.45}$$

where $\eta_{\mathrm{E\ pp}}$ denotes the net energy efficiency of power plant.

Substituting the relations (5.44) and (5.45) in Eq. (5.42) we get:

$$\eta_{\mathrm{E\ el\ CHP}} = \eta_{\mathrm{E\ pp}} \frac{\eta'_{\mathrm{et}}}{(1 - \varepsilon_{\mathrm{el}})\eta_{\mathrm{et}}} \tag{5.46}$$

The partial gross efficiency of heat production in cogeneration results from the equation:

$$\eta_{\mathrm{E\,h\,ec}} = \frac{\dot{Q}_{\mathrm{CHP}}}{\left(\dot{P}\,\mathrm{LHV}\right)_{\mathrm{CHP}} - N_{\mathrm{el\,CHP}} \dfrac{(1-\varepsilon_{\mathrm{el}})\eta_{\mathrm{et}}}{\eta_{\mathrm{E\,pp}}\eta'_{\mathrm{et}}}} \tag{5.47}$$

where

$\dot{Q}_{\mathrm{CHP}}$ production of heat flux in cogeneration,

$\left(\dot{P}\,\mathrm{LHV}\right)_{\mathrm{CHP}}$ consumption of the chemical energy of fuel in a CHP plant.

Equation (5.47) may be transformed as follows:

$$\eta_{\mathrm{E\,h\,CHP}} = \frac{\eta_{\mathrm{E\,CHP}}}{1-\sigma\left[\dfrac{\eta_{\mathrm{E\,CHP}}\eta_{\mathrm{et}}(1-\varepsilon_{\mathrm{el}})}{\eta_{\mathrm{E\,pp}}\eta'_{\mathrm{et}}}\right]} \tag{5.48}$$

$$\eta_{\mathrm{E\,CHP}} = \frac{\dot{Q}_{\mathrm{CHP}} + N_{\mathrm{el\,CHP}}}{\dot{P}LHV_{\mathrm{CHP}}} = \mathrm{EUF} \tag{5.49}$$

As $\eta_{\mathrm{E\,CHP}} > \eta_{\mathrm{E\,pp}}$, the partial efficiency of heat production in a CHP is higher than the total efficiency of the cogeneration process. The higher the values of the cogeneration index, the higher the values of the partial efficiency of heat production. Thus, it is energetically feasible to increase the parameters of live steam and decrease the pressure of the bleeding steam. The partial energy efficiency of heat production may attain values exceeding 1 (Fig. 5.15).

Such a result is physically correct because the CHP cycle results from the combined cycle of the heat engine with a heat pump whose energy efficiency is always higher than 1 [11].

5.4.4 Ecological Effects of Cogeneration

Savings in the chemical energy of fuels attained thanks to cogeneration result in a decreased emission of noxious substances. The emission of the *i*th noxious substance caused by the separate production of heat and electricity is:

$$S_{i\,s} = \frac{Q_0}{\eta'_{h\,t}\eta_{E\,hp}\mathrm{LHV}_{hp}} e_{i\,hp} + \frac{E_{el\,0}}{\eta'_{et}\eta_{E\,pp}\mathrm{LHV}_{pp}} e_{i\,pp} \tag{5.50}$$

where

$S_{i\,s}$ emission of the *i*th noxious substance in the case of separate production of heat and electricity,

$e_{i\,hp}$ index of the *i*th noxious substance emission in heating plant,

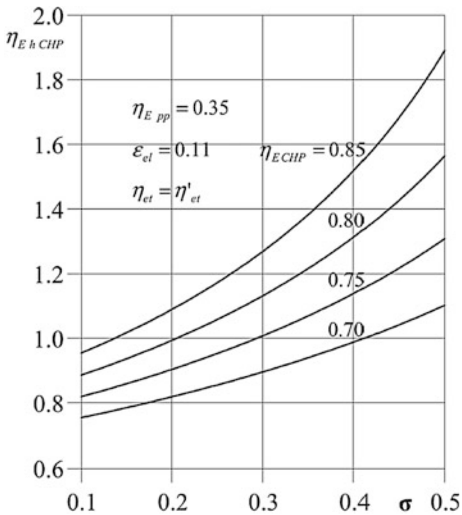


Fig. 5.15 Partial energy efficiency of heat production in a CHP plant

$e_{i\,pp}$ index of the ith noxious substance emission in power plant,
$\mathrm{LHV}_{hp}, \mathrm{LHV}_{pp}$ lower calorific value of fuel in heating plant and power plant, respectively.

The emission of the ith noxious substance in a CHP plant results from the relation:

$$S_{i\,\mathrm{CHP}} = \frac{1}{\eta_{\mathrm{E\,CHP}}\mathrm{LHV_{CHP}}}\left[\frac{Q_0}{\eta_{\mathrm{h\,t}}(1-\varepsilon_{\mathrm{h}})} + \frac{E_{\mathrm{el\,0}}}{\eta_{\mathrm{et}}(1-\varepsilon_{\mathrm{el}})}\right] e_{\mathrm{i\,CHP}} \tag{5.51}$$

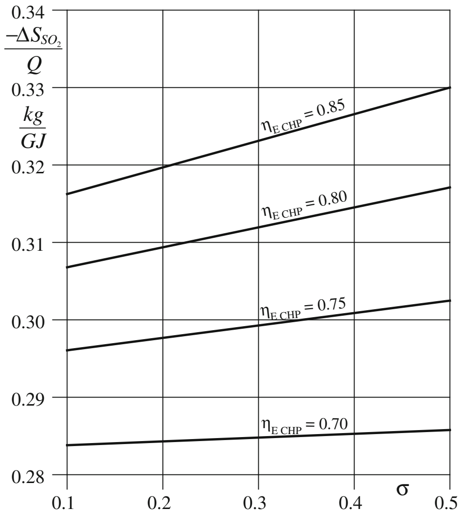
where
$e_{i\,\mathrm{CHP}}$ index of ith noxious substance emission in CHP,
$\mathrm{LHV_{CHP}}$ lower calorific value of fuel in CHP plant.

Assuming that the efficiencies of the transmission of heat and electricity are the same in the case of separate production of heat and electricity as in the CHP and that the LHV is the same, we obtain the following equation related to a unit of heat:

$$-\frac{\Delta S_i}{Q} = \frac{1}{\mathrm{LHV}}\left[\frac{e_{i\,hp}}{\eta_{\mathrm{E\,hp}}} - \frac{e_{\mathrm{i\,CHP}}}{\eta_{\mathrm{E\,CHP}}(1-\varepsilon_{\mathrm{h}})}\right](1-\varepsilon_{\mathrm{h}}) + \sigma\left[\frac{e_{\mathrm{i\,pp}}}{\eta_{\mathrm{E\,pp}}} - \frac{e_{\mathrm{i\,CHP}}}{\eta_{\mathrm{E\,CHP}}(1-\varepsilon_{\mathrm{CHP}})}\right](1-\varepsilon_{\mathrm{CHP}}) \tag{5.52}$$

The ecological effects of cogeneration depend not only on saving the chemical energy of fuel but also on the emission coefficients in both the separate production of heat and electricity and CHP plants. Figure 5.16 presents, for example, a reduced SO_2 emission.

Fig. 5.16 Reduction of SO2 emission



5.4.5 Realization of Cogeneration by Adapting the Power Unit to Heat Production [10, 14, 18]

The adaptation of power units for the production of heat leads to cogeneration and to savings in the chemical energy of fuels. Keeping the consumption of the chemical energy of fuel at a constant level, the production of electricity is reduced due to the production of heat. This decrease in electricity production can be compensated by the additional production of electricity in a replacing power station. Figure 5.17 presents a diagram of the separate production of heat and electricity: heating plant and power station. In Fig. 5.18, we have a diagram of a power unit adapted to the production of heat.

The consumption of chemical energy in the case of separate production is expressed by the equation:

$$E_{\mathrm{ch\ s}} = E_{\mathrm{ch\ pp}} + E_{\mathrm{ch\ hp}} \tag{5.53}$$

where

$E_{\mathrm{ch\ pp}}$ consumption of chemical energy of fuel concerning the power unit before adaptation,

$E_{\mathrm{ch\ hp}}$ consumption of chemical energy of fuel concerning the heating plant.

The consumption of the chemical energy of fuel in a cogeneration system looks like:

$$E_{\mathrm{ch\ cog}} = E_{\mathrm{ch\ pp}} + \Delta E_{\mathrm{ch\ r}} \tag{5.54}$$

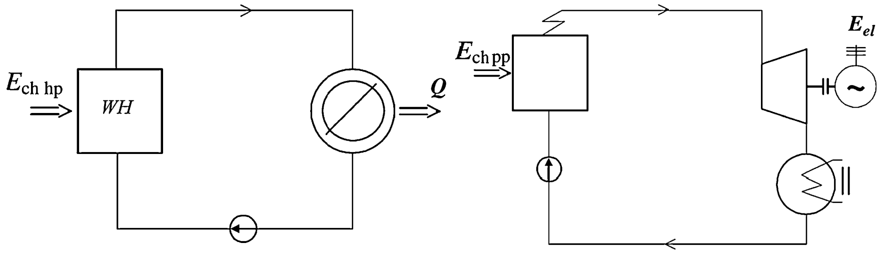


Fig. 5.17 Separate production of heat and electricity

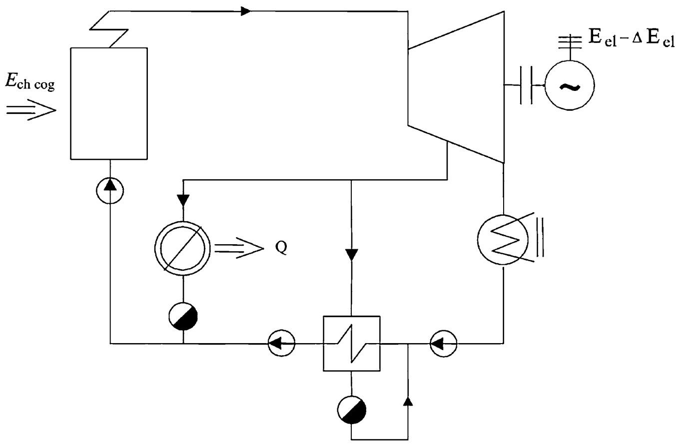


Fig. 5.18 Power plant after its adaptation to heat production; *WH* water heater

where $\Delta E_{ch\,r}$ denotes the consumption of chemical energy of fuel in a replacing power plant in which the decrease of electricity is compensated.

From Eqs. (5.53) and (5.54) we get the relation of savings $-\Delta E_{ch}$ of the chemical energy of fuels caused by the adaptation of a power unit to heat production:

$$-\Delta E_{ch} = \frac{Q_0}{\eta'_{ht}\eta_{E\,hp}} - \frac{\Delta E_{el}}{\eta_{E\,pp}} \tag{5.55}$$

where

Q_0	demand of heat (loco consumer),
η'_{ht}	energy efficiency of heat transmission in the case of a heating plant,
$\eta_{E\,hp}$	energy efficiency of the heating plant,
$\eta_{E\,pp}$	net energy efficiency of the replaced power plant,
ΔE_{el}	production of electricity in replaced power plant equal to a decrease in electricity production concerning power unit adapted for heat production.

In the analysis, the ratio of the reduction of electricity production to the amount of heat obtained from an adapted power unit is introduced. This relation is also called *coefficient of power decrease of the turbogenerator*:

$$u = \frac{-\Delta E_{\mathrm{el}}}{Q_0} \eta_{\mathrm{ht}} \tag{5.56}$$

where η_{ht} denotes the efficiency of heat transmission concerning the power plant adapted for heat production.

If Eq. (5.55) is transformed by dividing both sides by Q and applying relation (5.56) we get:

$$-\frac{\Delta E_{\mathrm{ch}}}{Q} = \frac{\eta_{\mathrm{ht}}}{\eta'_{\mathrm{ht}} \eta_{\mathrm{E\,hp}}} - \frac{u}{\eta_{\mathrm{E\,pp}}} \tag{5.57}$$

Figure 5.19 shows relation (5.57) as a function of the coefficient of power decrease assuming as a parameter the expression $\frac{\eta_{\mathrm{E\,hp}} \eta'_{\mathrm{ht}}}{\eta_{\mathrm{ht}}}$. Savings in the chemical energy of fuel achieved by adapting the power unit to the production of heat is higher, the lower the coefficient u, which means the lower the pressure of steam used for heating purposes and the lower the efficiency of the heating plant. If, for instance, the heating plant efficiency amounts to 0.85 and u = 0.15, we get $-\Delta E_{\mathrm{ch}}/Q = 0.7$, which means that at 100 MW of the heat flux we save about 70 MW of the flux of chemical energy of the fuels.

As a result of the adaptation of the power unit for heat production the electricity production drops and must be compensated by the production of electricity in the replacing power station. Thus, the production of heat in a power unit adapted for the production of heat ought to be charged at the same rate as the consumption of the chemical energy of fuel as in the replacing power station. Thus

$$\eta_{\mathrm{E\,h\,pp}} = \frac{Q}{\Delta E_{\mathrm{el}}} \eta_{\mathrm{E\,pp}} \tag{5.58}$$

or

$$\eta_{\mathrm{E\,h\,pp}} = \frac{\eta_{\mathrm{E\,pp}}}{u} \tag{5.59}$$

where $\eta_{\mathrm{E\,h\,pp}}$ denotes the partial efficiency of heat production in the power station adapted to heat production.

In the case of the most often encountered values u = 0.15 and $\eta_{\mathrm{E\,pp}} = 0.36$, the partial efficiency of heat production amounts to $\eta_{\mathrm{E\,h\,el}} = 2.4$. As mentioned above, this result is physically correct because of the idea of cogeneration. The lower the effect of decreasing the electricity production due to the production of heat, the higher the value of the partial efficiency of heat production.

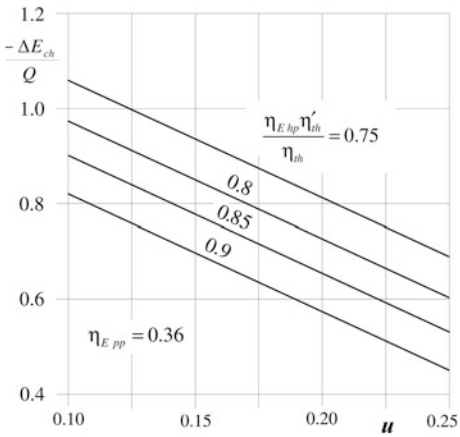


Fig. 5.19 Savings in the chemical energy of fuel due to adapting the power plant to the production of heat

Ecological Effects

In the chemical energy of fuel attained, thanks to the adaptation of a power unit for the production of heat cause a decrease in the emission of noxious substances into the environment:

$$-\Delta S_i = S_{i\,\mathrm{hp}} - \Delta S_{i\,\mathrm{pp}} \tag{5.60}$$

where

$S_{i\,\mathrm{hp}}$ emission of the *i*th noxious substance concerning the heating plant,

$\Delta S_{i\,\mathrm{pp}}$ increase of the *i*th noxious substance emission in the replacing power plant.

The emission of the *i*th noxious substance is in the case of a heating plant expressed by the relation:

$$S_{i\,\mathrm{hp}} = \frac{Q_0}{\eta_{\mathrm{ht}}\eta_{\mathrm{E\,hp}}\mathrm{LHV_{hp}}}\sigma_{i\,\mathrm{hp}} \tag{5.61}$$

where

$\sigma_{i\,\mathrm{hp}}$ unit emission of the *i*th noxious substance concerning heating plant,

$\mathrm{LHV_{hp}}$ lower calorific value of fuel in a heating plant.

The increase of the emission of the *i*th noxious substance in a replaced power plant results from the relation:

$$\Delta S_{i\,\mathrm{pp}} = \frac{\Delta E_{\mathrm{el}}}{\eta_{\mathrm{E\,pp}}\mathrm{LHV_{pp}}}\sigma_{i\,\mathrm{pp}} \tag{5.62}$$

where

$\sigma_{i\,\mathrm{pp}}$ unit emission of the *i*th noxious substance concerning power plant,

$\mathrm{LHV_{pp}}$ lower calorific value of fuel in power plant.

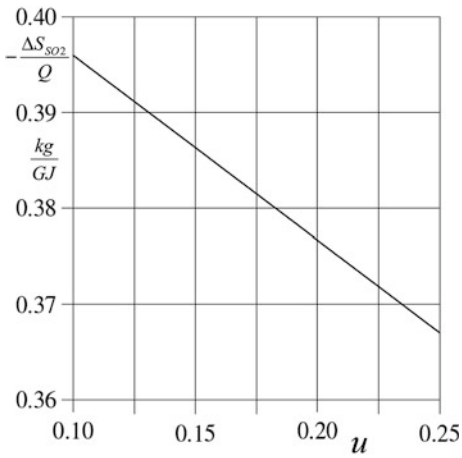


Fig. 5.20 Decrease in SO_2 emission due to adapting the power plant to the production of heat

Substituting Eqs. (5.61) and (5.62) into Eq. (5.60) and taking into account relation (5.56), and dividing both sides by $Q = Q_0/\eta_{\mathrm{ht}}$ we get:

$$-\frac{\Delta S_i}{Q} = \frac{\eta_{\mathrm{ht}}\sigma_{i\,\mathrm{hp}}}{\eta'_{\mathrm{ht}}\eta_{\mathrm{E\,hp}}\mathrm{LHV}_{\mathrm{hp}}} - \varepsilon\frac{\sigma_{i\,\mathrm{pp}}}{\eta_{E\,\mathrm{pp}}\mathrm{LHV}_{\mathrm{pp}}} \tag{5.63}$$

Assuming the following data: $\sigma_{\mathrm{SO_2\,hp}} = 8.5$ g/kg fuel; $\sigma_{\mathrm{SO_2\,pp}} = 1.6$ g/kg fuel; $\mathrm{LHV}_{\mathrm{hp}} = \mathrm{LHV}_{\mathrm{pp}} = 23$ MJ/kg; $\eta_{\mathrm{ht}} = 0.89$; $\eta'_{\mathrm{ht}} = 0.85$; $\eta_{\mathrm{E\,hp}} = 0.85$; $\eta_{\mathrm{E\,pp}} = 0.36$ a diagram of the relation $-\Delta SO_2/Q$ as a function of u has been plotted (Fig 5.20).

5.4.6 Gas and Gas-and-Steam CHP Units

A simple gas CHP unit consists of a gas turbine and a heat recovery boiler used to heat the network water (Fig. 5.21) [10]. The combustion chamber is fed with compressed natural gas from the gas grid and with compressed air. The energy efficiency of the heat recovery boiler is high because the temperature of the flue gases leaving the heat recovery boiler may be only slightly higher than the temperature of the network return water. The exergy efficiency of the heat recovery boiler is, however, much lower due to the considerable difference between the temperature of the flue gases and the temperature of the network water which leads to considerable losses of exergy. The application of flue gas for the purpose of preheating network water considerably improves the energy efficiency of the cogeneration system if compared with the open cycle of the gas turbine, but the improvement in exergy efficiency is much lower.

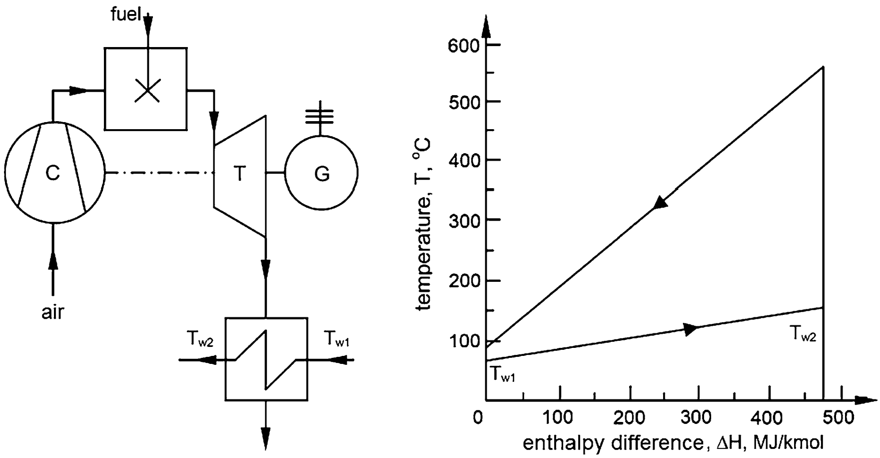


Fig. 5.21 Scheme of a simple gas CHP unit and temperature distribution in heat recovery boiler [10]; *C* compressor; *T* gas turbine, *G* generator; T_{w1}, T_{w2} temperature of network water

$$\eta_{\mathrm{B\,CHP}} = \frac{E_{\mathrm{el}} + Q\frac{T_m - T_a}{T_m}}{a\,\mathrm{LHV}} = \eta_{\mathrm{B\,g}} + \Delta\eta_{\mathrm{B}} \tag{5.64}$$

$$\Delta\eta_{\mathrm{B}} = \frac{Q}{a\,\mathrm{LHV}} \cdot \frac{T_m - T_a}{T_m} \tag{5.65}$$

where

$\eta_{\mathrm{B\,CHP}}$	exergy efficiency of CHP unit,
E_{el}	production of electricity,
Q	production of heat,
T_m	thermodynamic average temperature (Appendix C),
T_a	ambient temperature,
a	ratio of chemical exergy of fuel to LHV,
LHV	lower heating value,
$\eta_{\mathrm{B\,g}}$	electrical efficiency of gas turbine cycle,
$\Delta\eta_{\mathrm{B}}$	increase in exergy efficiency thanks to preheating of network water.

The exergy efficiency of a simple gas CHP unit can be improved by preheating the compressed air by high-temperature flue gases leaving the gas turbine. Only after leaving the preheater of compressed air the flue gases are used to preheat the network water. This solution is, however, much more expensive due to the high expenditure needed for the high-temperature preheater of compressed air. Another way of improving the exergy efficiency of a gas CHP unit is to utilize high-temperature flue gases for so-called chemical recuperation consisting in the application of flue gases for the conversion of methane contained in natural gas. As the exergy of flue gases is applied for the production of heat only in the range of a

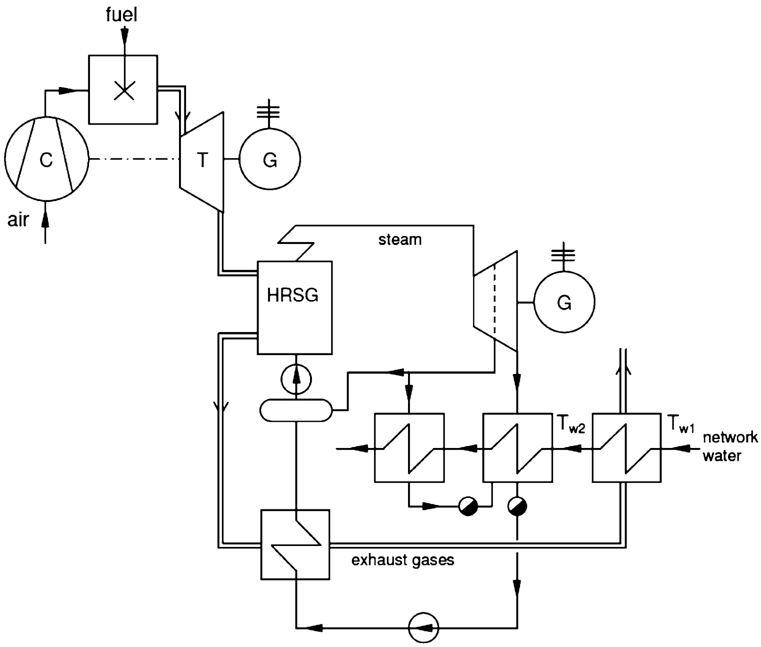


Fig. 5.22 Single-pressure gas and steam CHP unit; *C* compressor; *T* gas turbine, *G* generator; T_{w1}, T_{w2} temperature of network water; *HRSG* heat recovery steam generator

reduced temperature of the flue gases, the ratio of the electrical power to the heating power is in this system higher than in a simple gas CHP [10].

The exergy efficiency of a simple gas CHP unit can be improved by utilizing the high-temperature flue gases leaving the turbine for the production of steam in the heat recovery steam generator, and for driving the steam turbine. Figures 5.22 and 5.23 [10] present the scheme of a single-pressure steam-and-gas CHP unit and temperature distribution in the heat recovery steam generator. A drawback of this solution is that there are considerable exergy losses in the evaporator. Therefore, this solution is usually realized only in small-scale CHP plants (e.g., in complex buildings). The ratio of the electrical power to the heating power may reach the value of about 1, whereas in steam CHP plants it amounts to 0.45.

Gas-and-steam systems combine the advantages of both separate gas and steam cycles eliminating their drawbacks [4, 8, 10]. The gas cycle (cycle of the gas turbine) displays the advantage that a much higher temperature of the working fluid can be applied thanks to direct heat transfer. A drawback of the gas cycle, however, is the high temperature of the flue gas at the outlet from the gas turbine. A drawback of the steam cycle is rather low parameters of the live steam. Due to material limitations in the steam cycle the considerable difference in temperature between combustion gas and the working fluid is not sufficiently utilized, the consequence of which are high exergy losses. An advantage of the system cycle is a low temperature of condensation (low isotherm of the thermodynamic cycle of

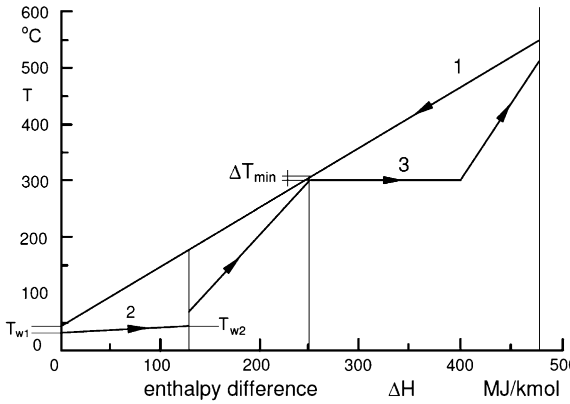


Fig. 5.23 Temperature distribution in the single-pressure heat recovery steam generator; *1* combustion gases, *2* network water, *3* water in the steam cycle, ΔT_{min} pinch point temperature difference

the engine). The advantages of both cycles, connected with a simultaneous elimination of their drawbacks, mean that the energy efficiency of the combined gas and steam cycle amounts to 50–55 (60 %). The application of a gas-and-steam cycle is also ecologically beneficial because emissions of SO_2, NO_x, CO_2, and dust are reduced. An additional advantageous effect is lower demand for industrial water and reduction in irreclaimable losses of water. The gas-and-steam cycle belongs to the group of cascade processes obtained by connecting the cycle of various working media characterized by convenient properties in different ranges of temperature. Additional energy-ecological effects are achieved by applying gas-and-steam cycles in CHP plants. Gas-and-steam CHP plants fired with natural gas ensure higher savings in the chemical energy of fuels than those achieved in CHP plants fired with hard coal, due to the higher index of cogeneration.

The following ways of joining the gas cycle with steam cycle, realized in practice, can be distinguished as:

- series connection in which the input energy is supplied to the combustion chamber of the gas turbine, and the steam generator situated behind the outlet of the gas turbine plays the role of a heat recovery boiler,
- series–parallel connections in which the input energy is supplied to both the combustion chamber of the gas turbine and the steam generator, to which the flue gases from the outlet of the gas turbine are supplied containing a large amount of oxygen; this system is often called an after-burning system,
- parallel connections in which the steam cycle is fed from two independent units, viz., the steam boiler and the heat recovery boiler of the gas cycle; both these units are joined only on the steam side.

Series connections are applied mainly in recently designed gas-and-steam systems. Such a connection is most useful from the thermodynamic point of view because the advantages of the gas cycle are fully utilized. The energy efficiency of the gas-and-steam cycle depends, first of all, on the difference of the temperature between the combustion gases and the temperature of water and steam in the heat

recovery boiler. The application of double- or three-pressure heat recovery steam generator results in a decrease of exergy losses and an improvement in the thermodynamic performance of the cycle.

In the gas turbine system the excess air ratio in the combustion chamber is higher in order to reduce the temperature of flue gases before entering the gas turbine. This, however, lowers the efficiency of the gas turbine because the excess air is irreversibly compressed and then also irreversibly expanded (as a component of the flue gases), and this effect is accompanied by considerable losses of exergy. These losses may be reduced by injecting steam into the working medium before it is pressed into the gas turbine (so-called Cheng cycle) or by injecting water into the compressed air (humid air turbine, HAT). In the latter case, it should be stressed that the compression of water requires much less input energy than the compression of air. HAT provides good conditions for the production of heat in cogeneration. The flue gases leaving the turbine are characterized by a temperature of about 120 °C and contain much steam. Thanks to its condensation much heat can be obtained for preheating network water without considerable exergy losses.

The construction of small-scale CHP units at the place of the consumption of heat permits to considerably reduce the costs of heating networks and losses in transporting the heat. The construction of these CHP units is, however, expedient when:

- demand for heat is not too small,
- the CHP is automatized and easily handled,
- its operation is not strenuous for the consumers (due to noise, vibrations, the emission of flue gases, etc.),
- the cost of fuels is not too high.

Small-scale CHP units are equipped with piston combustion engines or gas turbines. The former may be fed with liquid or gaseous fuels(e.g., natural gas or biogas). A single piston engine as a module of the CHP unit has a power rating of 70–300 kW and a heating flux of 130–550 kW. Development in the construction of small-scale gas turbines has made it possible to realize units below 70 kW (even of 30 kW).

The electrical efficiencies of a piston combustion engine and small-scale gas turbine are comparable (34–37 %). Also, the energy efficiency of the whole CHP units is in both cases similar. Better energy effects are achieved in gas-and-steam cycles if the heat recovery steam generator, fed with combustion gases from the piston engine or gas turbine, serves to produce steam for driving the steam turbine, cogenerating heat and electricity. In small-scale gas turbines more and more often regeneration of heat is included, i.e., the compressed air is preheated by means of flue gases, which distinctly improves the efficiency of the turbine although it lowers the temperature of the flue gases before entering the heat recovery steam generator.

The cogeneration of heat and electricity in small-scale CHP units can be conveniently realized by applying combustion piston engines. In this type of CHP units heat is received from the cooling system of the engine and the combustion

gases leaving the engine. A CHP plant with combustion piston engine is usually constructed modularly. The respective modules can be thrown into and out of gear. This system is usually supplemented by a water heater.

The heat obtained from the cooling system of the engine and from the flux of combustion gases at the outlet of turbine may be expressed by the formula [10]:

$$\dot{Q}_u = \dot{P}\,\mathrm{LHV}\left(\varepsilon_\mathrm{c} + \varepsilon_\mathrm{fg}\eta_\mathrm{E\,sg}\right) \tag{5.66}$$

where

$\dot{Q}_u$	useful heat flux,
$\dot{P}$	consumption of fuel,
$\varepsilon_\mathrm{c},\ \varepsilon_\mathrm{fg}$	relative energy losses concerning the cooling system and outlet of flue gases,
$\eta_\mathrm{E\,sg}$	energy efficiency of heat recovery steam generator.

The electrical power is calculated as follows:

$$N_\mathrm{el} = \dot{Q}_u \frac{\eta_\mathrm{E\,pce}}{\varepsilon_\mathrm{c} + \varepsilon_\mathrm{fg}\eta_\mathrm{E\,sg}} \tag{5.67}$$

where $\eta_\mathrm{E\,pce}$ denotes electrical energy efficiency of combustion engine.

Savings in the cumulative energy of fuels in the case of gas and gas-and-steam CHP plants are calculated by means of the relation:

$$-\Delta E^*_\mathrm{ch} = \frac{N_\mathrm{el}\eta_\mathrm{et}}{\eta_\mathrm{E\,pp}\eta^*_\mathrm{d\,pp}\eta'_\mathrm{et}} + \frac{\dot{Q}\eta_\mathrm{ht}}{\eta_\mathrm{E\,hp}\eta^*_\mathrm{d\,hp}\eta'_\mathrm{ht}} - \frac{\dot{P}\mathrm{LHV}}{\eta^*_\mathrm{d\,CHP}} \tag{5.68}$$

where

$-\Delta E^*_\mathrm{ch}$	savings in the cumulative energy of fuels achieved by cogeneration,
N_el	electric power produced in the CHP,
$\eta_\mathrm{et},\ \eta'_\mathrm{et}$	efficiencies of the transformation and transmission of electricity from the CHP plant and the replaced power station,
$\eta_\mathrm{E\,pp}$	net energy efficiency of the replaced power plant,
$\eta^*_\mathrm{d\,pp},\ \eta^*_\mathrm{d\,hp},\ \eta^*_\mathrm{d\,CHP}$	cumulated energy efficiencies of delivering the fuel to the replaced power station, the heating plant, and the gas-and-steam CHP plant,
$\dot{Q}$	thermal power of the CHP plant,
$\eta_\mathrm{ht},\ \eta'_\mathrm{ht}$	efficiency of transmitting heat from the CHP plant and the replaced heating plant to the consumers,
$\dot{P}\mathrm{LHV}$	consumption of the chemical energy of fuel in a gas or gas-and-steam CHP plants.

The choice of the replaced power station should depend on the ability of the CHP plant to produce electricity in the respective zones of the load of the domestic

electro-energy system. The least effective case results from the assumption of replacing electricity produced in the basic zone. Then, the efficiency of the replaced power station concerns the most modern power station.

5.5 Trigeneration: CHP Plant Integrated with a Cooling System

5.5.1 Introduction

Similarly, as in the case of centralized supply of heat, which has a long tradition, the centralized production and transmission of the cooling agents is possible. Cooling centers are usually applied in both industrial plants and in large complex buildings. The cooling agent can be produced by means of compression or absorption cooling units. In the case of CHP plants integrated with cooling systems, the application of absorption coolers is of special interest.

Demands for heat and cooling agents change mutually in the course of a year. Figure 5.24 presents as an example the actual annual diagram of heat and a cooling agent demand [17]. In summer, the demand for a cooling agent is covered by systems with absorption coolers, influencing the energy and economic effects of CHP. Thus, so-called “trigeneration” systems come into play, that is, heat and electricity cogeneration is integrated with the production of the cooling agent. Trigeneration systems can also be realized by applying compression coolers or hybrid systems. Trigeneration systems applied in complex buildings are called BCHP (Buildings, Cooling, Heating, and Power). The crucial elements in this technique are absorption coolers, which make it possible to utilize waste heat for the purpose of cooling.

5.5.2 Centralized Production of Cooling Agents

As mentioned in the introduction to this section, similarly as in centralized heat production there are also systems of centralized production of the cooling agents. In the latter case, the temperature of the feeding water amounts to 4–5 °C and the temperature of the returned water to 12–15 °C. Generally, the difference between the temperature of the feeding and returned water amounts to 9 K, and the applied parameters are 4/13 °C [7].

In centralized cooling systems compression or absorption coolers may be used. If the price of electricity is favorable (e.g., electricity from hydro-electric power stations) it is feasible to apply compression coolers. When the structure of electricity production is dominated by thermal power plants, the application of absorption coolers provided with heat from CHP plants is energetically and economically more effective. There are also systems of centralized production of

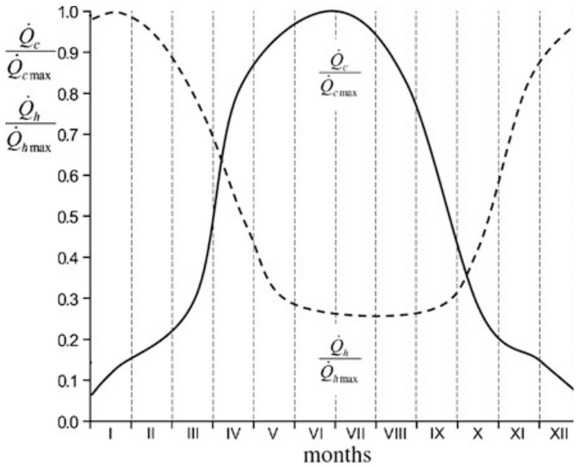


Fig. 5.24 Exemplary annual diagram of heat and cooling agent demand adapted from Zahoransky [17]

cooling agents fed by heat from heating plants or a source of waste heat. Where an absorption cooler is used with a CHP unit provided with gas and steam cycle, input energy carriers may be flue gases, heating steam, or hot water.

In comparison with individual systems a centralized supply with cooling agents is more effective thanks to the improved efficiency of the production of the cooling agent, the lower costs of operation, and a smaller need of site. In choosing a centralized delivery of cooling agents, however, the location of the cooling center must be taken into account, as well as the kind and cost of input energy and the possibility of applying differentiated tariffs for electricity.

The effectiveness of centralized cooling systems can be increased by applying cooling storage tanks, which permit the operation of cooling installations to be adapted to partial loading, a reduction in the capital cost thanks to a lower power rating of the cooler, and also a reduction in the cost of electricity (if the accumulator can be loaded at a time when the tariff is more favorable) [7].

5.5.3 Trigeneration

As already mentioned, a CHP unit with a centralized system of providing the cooling agent is called trigeneration and the cooling system may be provided with either a compressor cooler or an absorption cooler. A series system is also possible in which the water is cooled down first in the absorption cooler and then in the compressor cooler. Further considerations are devoted to the analysis of a centralized cooling system with an absorption cooler Fig. (5.25).

Savings in the chemical energy of the fuel achieved thanks to the trigeneration system, compared with a separate system producing heat, cooling agents, and electricity result from the following equation:

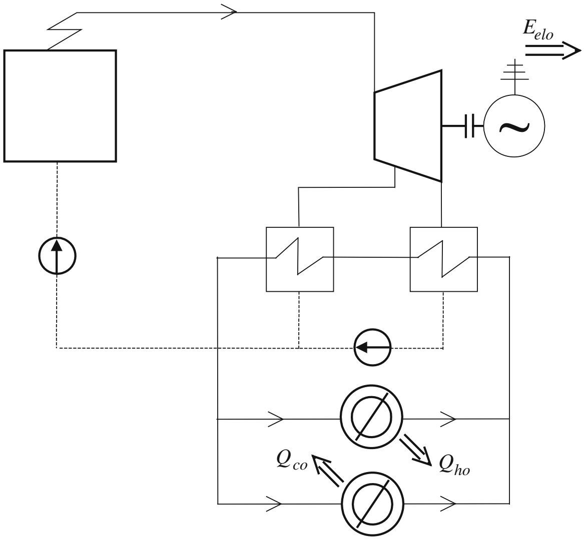


Fig. 5.25 Scheme of a CHP unit with a centralized system of providing the cooling agent

$$-\Delta E_{\mathrm{ch\,t}} = Q_{\mathrm{h0}}\left[\frac{1}{\eta_{\mathrm{E\,hp}}\eta'_{\mathrm{ht}}} - \frac{1}{\eta_{\mathrm{E\,CHP}}\eta_{\mathrm{ht}}(1-\varepsilon_{\mathrm{h}})}\right] + \frac{Q_{\mathrm{co}}}{\mathrm{COP}_a}\left[\frac{1}{\eta_{\mathrm{E\,hp}}\eta'_{\mathrm{c\,t}}} - \frac{1}{\eta_{\mathrm{E\,CHP}}\eta_{\mathrm{ct}}(1-\varepsilon_c)}\right] + $$
$$+ E_{\mathrm{el}\,o}\left[\frac{1}{\eta_{\mathrm{E\,pp}}\eta'_{\mathrm{et}}} - \frac{1}{\eta_{\mathrm{E\,CHP}}\eta_{\mathrm{et}}(1-\varepsilon_{\mathrm{el}})}\right] \tag{5.69}$$

where

$-\Delta E_{\mathrm{ch\,t}}$	saving of the chemical energy of fuel in the case of trigeneration,
Q_{ho}	heat demand of loco consumer,
Q_{co}	cooling agent demand of loco consumer,
E_{elo}	electricity demand of loco consumer,
$\eta_{\mathrm{E\,hp}}$	net energy efficiency of the heating plant,
$\eta_{\mathrm{E\,CHP}}$	gross energy efficiency of the CHP plant,
$\eta_{\mathrm{ht}}, \eta'_{\mathrm{ht}}$	efficiency of heat transmission from CHP and heating plant, respectively,
η_{Epp}	net energy efficiency of the power plant,
$\eta_{\mathrm{ct}}, \eta'_{\mathrm{ct}}$	efficiency of transmission of the cooling agent from the trigeneration system and from separate installation,
COP_a	coefficient of performance of the absorption chiller,
ε_{h}	coefficient of own heat consumption,
ε_{c}	coefficient of own cooling agent consumption,
$\varepsilon_{\mathrm{el}}$	coefficient of own electricity consumption in the CHP.

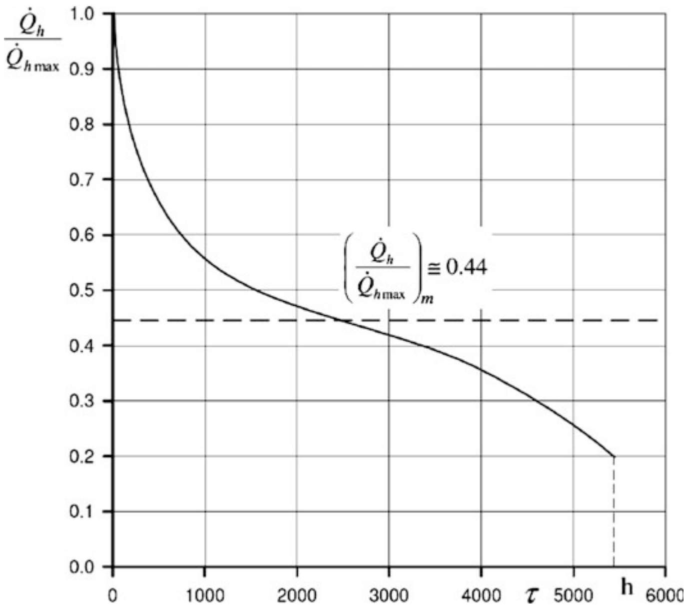


Fig. 5.26 Duration curve of heat demand

In relation (5.69) it is assumed that the demand for heat, cooling agent, and electricity is the same in the trigeneration system as it is in the separate production of the aforementioned energy carriers. The consumption of electricity in the absorption cooler has been neglected because it is inconsiderable.

Figures 5.26 and 5.27 present reduced duration curves of heat in the heating season and of the cooling agent in summer. The duration of heating is assumed to be $\tau_{\mathrm{h}} = 5{,}400\,\mathrm{h}$, and the duration of the demand for the cooling agent $\tau_{\mathrm{c}} = 1{,}800\,\mathrm{h}$ [7].

The annual demand for heat and cooling agents by consumers is expressed by the relations:

$$Q_{\mathrm{ho}} \cong 0{,}44\,\dot{Q}_{\mathrm{h\ max}\ o}\tau_{\mathrm{h}} \tag{5.70}$$

$$Q_{\mathrm{co}} \cong 0{,}42\,\dot{Q}_{\mathrm{c\ max}\ o}\tau_{\mathrm{c}} \tag{5.71}$$

where

$Q_{\mathrm{h}o}$	annual demand for heat,
$\dot{Q}_{\mathrm{h\ max}\ o}$	maximum flux of heat loco consumer,
τ_{h}	duration of the heating season,
Q_{c0}	annual demand for cooling agent,
$\dot{Q}_{\mathrm{c\ max}\ o}$	maximum flux of the cooling agent loco consumer,
τ_{c}	duration of the demand for the cooling agent.

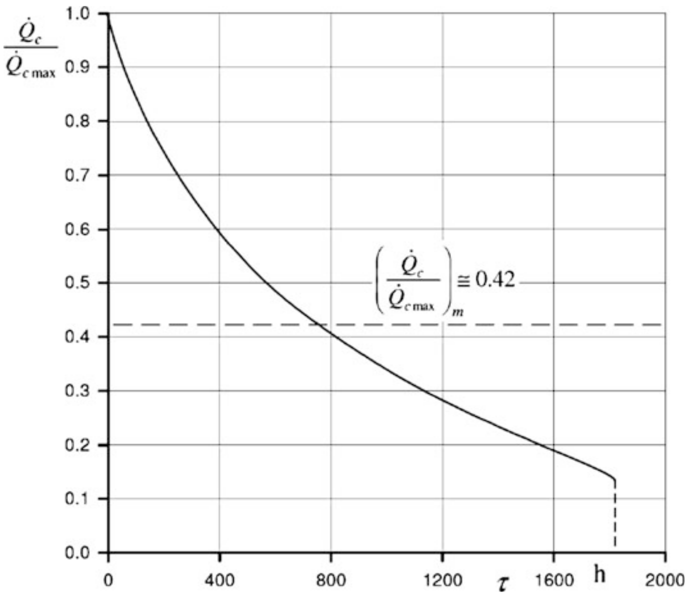


Fig. 5.27 Duration curve of cooling agent demand

The ratio of the demand for cooling agents to the demand for heat may be expressed as:

$$\frac{Q_{co}}{Q_{ho}} \cong \frac{1}{3}\frac{\dot{Q}_{c\ max\ o}}{\dot{Q}_{h\ max\ o}} \tag{5.72}$$

Denoting the ratio Q_{co}/Q_{ho} by α, the relation between the gross production of the cooling agent and heat results from the equation:

$$Q_c = \alpha\frac{\eta_{ht}(1-\varepsilon_h)}{\eta_{ct}(1-\varepsilon_c)}Q_h \tag{5.73}$$

The relation between the production of heat and the cooling agent and the production of electricity is:

$$E_{el} = \sigma\left(Q_h + \frac{Q_c}{COP_a}\right) \tag{5.74}$$

where σ denotes the index of cogeneration.

Making use of Eqs. (5.72), (5.73), and (5.74) in Eq. (5.69), and assuming that, $\eta_{et} \cong \eta'_{et};\ \eta_{ht} \cong \eta'_{ht};\ \eta_{ct} \cong \eta'_{ct}; (1-\varepsilon_h) \cong (1-\varepsilon_c)$ we get [20]:

$$\left(-\frac{\Delta E_{ch}}{Q_h}\right)_t = \left(1+\frac{\alpha}{COP_a}\right)\left[\sigma\left(\frac{1-\varepsilon_{el}}{\eta_{E\ pp}} - \frac{1}{\eta_{E\ CHP}}\right) + \frac{1-\varepsilon_h}{\eta_{E\ hp}} - \frac{1}{\eta_{E\ CHP}}\right] \tag{5.75}$$

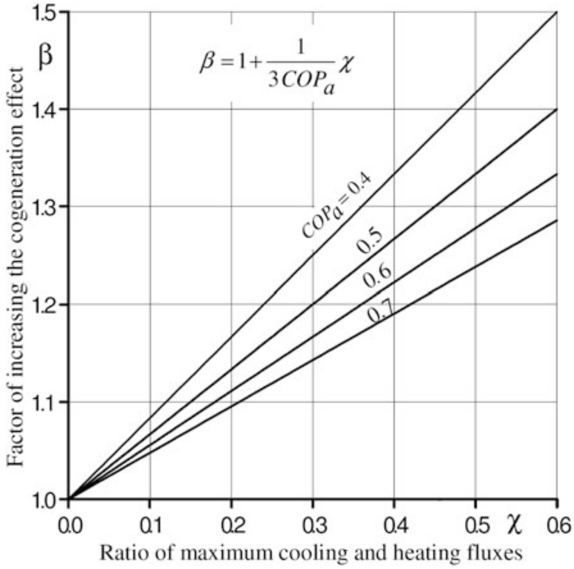


Fig. 5.28 Factor of increasing the cogeneration effect thanks to trigeneration

The effect of trigeneration is assessed quantitatively by the coefficient of the increase of cogeneration effects [20]:

$$\beta = \frac{\left(-\frac{\Delta E_{ch}}{Q_h}\right)_t}{-\frac{\Delta E_{ch}}{Q_h}} \tag{5.76}$$

Comparing (5.40) with (5.75) and applying (5.72) as well assuming that $\eta_{ht} = \eta_{ct}$, we get:

$$\beta \cong 1 + \frac{\chi}{3\,\mathrm{COP}_a} \tag{5.77}$$

$$\chi = \frac{\dot{Q}_{c\ \max\ o}}{\dot{Q}_{h\ \max\ o}} \tag{5.78}$$

The effect of increased savings in the chemical energy of fuel in the cogeneration system due to trigeneration is illustrated in Fig. 5.28 [20]. The results presented there must be interpreted as quantities calculated in comparison with separate production of heat, electricity, and cooling agent. This means that in the case of the given absorption cooler the effect of cogeneration grows with the increasing demand for the cooling agent. The growing values of the coefficient β with the decreasing $\mathrm{COP_a}$ (coefficient of performance) of the cooler should not be interpreted as meaning that the worse the $\mathrm{COP_a}$, the better, because the consumption of input energy in the cooler increases. We may only speak of less severe

consequences of the worse COP_a of the absorption cooler if it is integrated with a CHP plant. Thus, a CHP partially compensates the worse COP_a of the absorption chiller.

5.6 Analysis of the Index of Primary Energy Savings Concerning Cogeneration

The Directive 2004/8/EC [1, 2], a part of which has been quoted in the Appendix E to this book, includes the index PES (Primary Energy Savings), which determines relative savings in the chemical energy of fuels achieved, thanks to cogeneration. The equation (E2) in Appendix E defines the index PES. The term "reference energy efficiency" complies with the term "energy efficiency of a replaced process," up to now used in the literature [11, 12].

Substituting in Eq. (E1) terms expressing arithmetical partial efficiencies concerning the production of heat and electricity in cogeneration, viz. [19]:

$$\text{CHP } H\eta = \frac{Q_{\text{cog}}}{E_{\text{ch cog}}} \tag{5.79}$$

$$\text{CHP } E\eta = \frac{E_{\text{el cog}}}{E_{\text{ch cog}}} \tag{5.80}$$

we get

$$\text{PES} = 1 - \frac{E_{\text{ch cog}}}{E_{\text{ch hp}} + E_{\text{ch pp}}} = \frac{-\Delta E_{\text{ch}}}{E_{\text{ch sep}}} \tag{5.81}$$

where

$Q_{\text{cog}}, E_{\text{el cog}}$	production of heat and electricity in the CHP,
$E_{\text{ch cog}}$	consumption of the chemical energy of fuels in the CHP,
$E_{\text{ch hp}}, E_{\text{ch pp}}$	consumption of the chemical energy of fuels in a reference heating plant (replaced process) and reference power station (replaced process),
$-\Delta E_{\text{ch}}$	savings of the chemical energy of fuels,
$E_{\text{ch sep}}$	consumption of the chemical energy of fuels concerning separate production of heat and electricity.

Figure 5.11 presents a diagram of a CHP plant with a back-pressure turbine fired with coal. The consumption of the chemical energy of fuels in this CHP plant and in the separate production of heat and electricity is expressed by the following equations:

$$E_{\text{ch cog}} = \frac{(1+\sigma)Q_{\text{cog}}}{\eta_{\text{E CHP}}} \tag{5.82}$$

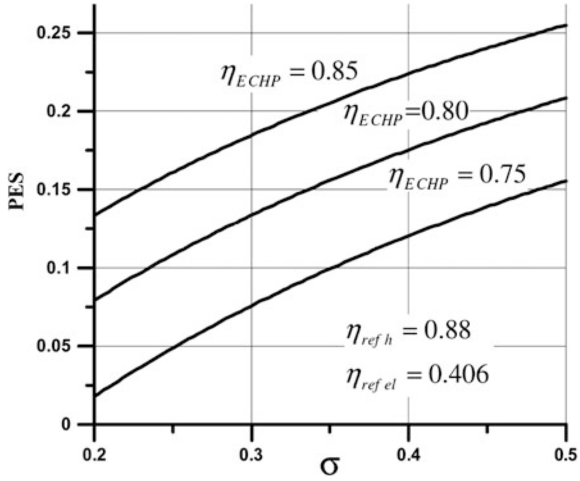


Fig. 5.29 Index PES in the case of a back-pressure turbine

$$E_{\mathrm{ch\,sep}} = Q_{\mathrm{cog}}\left(\frac{1}{\eta_{\mathrm{ref\,hp}}} + \frac{\sigma}{\eta_{\mathrm{ref\,pp}}}\right) \tag{5.83}$$

Substituting Eqs. (5.82) and (5.83) in (5.81) we get:

$$\mathrm{PES} = 1 - \frac{1+\sigma}{\eta_{\mathrm{E\,cog}}\left(\frac{1}{\eta_{\mathrm{ref\,hp}}} + \frac{\sigma}{\eta_{\mathrm{ref\,pp}}}\right)} \tag{5.84}$$

Figure 5.29 presents the index PES in the considered case, which shows that the condition PES $\geq$ 10 % might be safely satisfied by CHP plants with an efficiency exceeding 75 % and a index of cogeneration higher than 0.4.

Figure 5.18 presents a diagram of a power plant adapted for the production of heat. In this adaptation, it is assumed that the consumption of the chemical energy of fuels E_{ch} does not undergo any changes, viz.:

$$E_{\mathrm{ch}} = \frac{E_{\mathrm{el}}}{\eta_{\mathrm{E\,pp}}} = \mathrm{const} \tag{5.85}$$

The consumption of the chemical energy of fuels $E_{\mathrm{ch\,cog}}$, charging the production in cogeneration, results from the relation:

$$E_{\mathrm{ch\,cog}} = E_{\mathrm{ch}} - E_{\mathrm{ch\,c}} \tag{5.86}$$

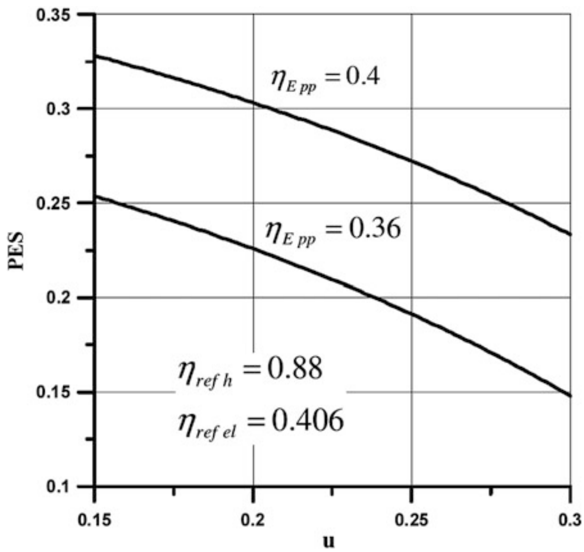


Fig. 5.30 Index PES concerning the power plant adapted for heat production

where

$$E_{\mathrm{chc}} = \frac{E_{\mathrm{elc}}}{\eta_{\mathrm{Epp}}} \tag{5.87}$$

where

$E_{\mathrm{el\,c}}$ production of electricity in the condensation mode after the adaptation of the power plant,

$\eta_{\mathrm{E\,pp}}$ energy efficiency of the production of electricity in the condensation mode.

The amount of electricity produced in the condensation mode after the adaptation of the power plant is calculated taking into account the index of power loss and the index of cogeneration:

$$E_{\mathrm{el\,c}} = E_{\mathrm{el}} - uQ_{\mathrm{cog}} - \sigma\, Q_{\mathrm{cog}} \tag{5.88}$$

where the coefficient of power decrease is expressed by:

$$u = \frac{-\Delta N_{\mathrm{el}}}{\dot{Q}_{\mathrm{cog}}} \tag{5.89}$$

where

$-\Delta N_{\mathrm{el}}$ power decrease due to adaptation,

$\dot{Q}_{\mathrm{cog}}$ flux of heat provided by the adapted power plant.

Introducing Eqs. (5.85), (5.87), and (5.88) in (5.86) we get:

$$E_{\mathrm{ch\,cog}} = \frac{(u + \sigma)Q_{\mathrm{cog}}}{\eta_{\mathrm{E\,pp}}} \tag{5.90}$$

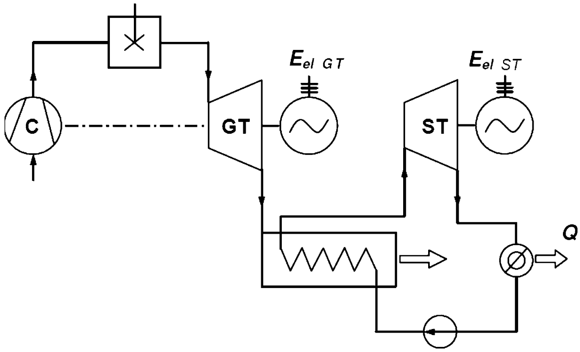


Fig. 5.31 Diagram of a gas-and-steam CHP plant; *C* compressor, *GT* gas turbine, *ST* steam turbine

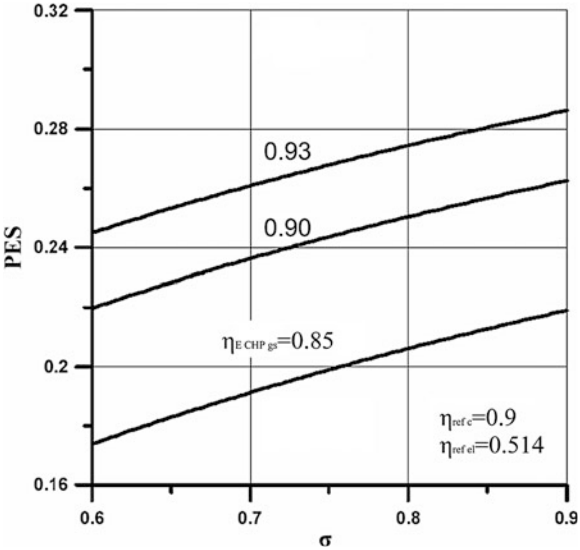


Fig. 5.32 Index PES concerning a gas-and-steam CHP plant

Hence:

$$\mathrm{PES} = 1 - \frac{\sigma + u}{\eta_{E\,pp}\left(\dfrac{1}{\eta_{ref\,hp}} + \dfrac{\sigma}{\eta_{ref\,pp}}\right)} \tag{5.91}$$

Figure 5.30 presents the index PES concerning the adapted power plant for heat production. The index PES depends strongly on the coefficient of power decrease. Consuming steam for heating purposes from the receiver between the medium- and low-pressure part of the turbine ($u \cong 0.2$), the index PES exceeds 0.2. The consumption of steam from the outlet of high-pressure part of the turbine ($u \cong 0.4$), however, leads to a drop in the index PES to less than 10 %.

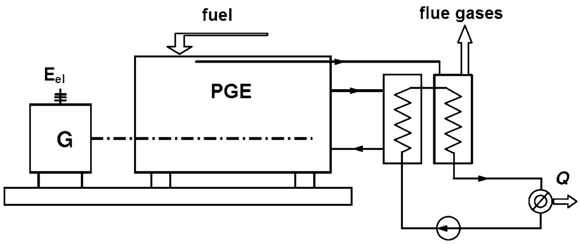


Fig. 5.33 Diagram of a small-scale CHP plant equipped with a piston gas engine; *PGE* piston gas engine; *G* generator

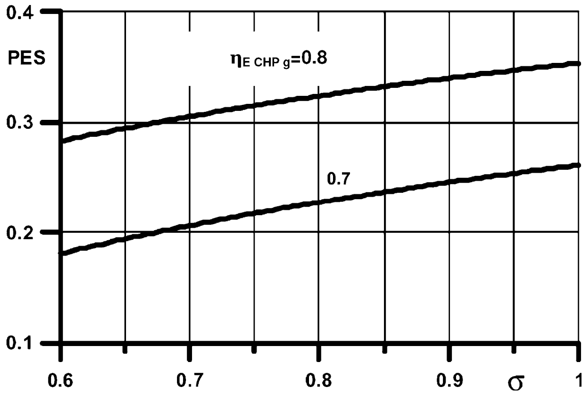


Fig. 5.34 Index PES concerning a small-scale CHP plant equipped with a piston gas engine

Figure 5.31 presents the diagram of a gas-and-steam CHP plant fired with natural gas. The index PES is calculated based on the relation (5.81). The consumption of the chemical energy of fuels is calculated based on the energy efficiency of the gas-and-steam CHP plant (Fig. 5.31).

$$E_{\mathrm{ch\,cog}} = \frac{Q_{\mathrm{cog}} + E_{\mathrm{el\,cog}}}{\eta_{\mathrm{E\,CHP\,gs}}} \tag{5.92}$$

The amount of electricity produced in cogeneration is calculated as follows:

$$E_{\mathrm{el\,cog}} = \sigma Q_{\mathrm{cog}} \tag{5.93}$$

Hence:

$$\mathrm{PES} = 1 - \frac{1+\sigma}{\eta_{\mathrm{E\,CHP\,gs}}\left(\frac{1}{\eta_{\mathrm{ref\,hp}}} + \frac{\sigma}{\eta_{\mathrm{ref\,pp}}}\right)} \tag{5.94}$$

Figure 5.32 presents the index PES, which indicates a high effectivity in the realization of the gas-and-steam CHP plant. In the case of modern gas-and-steam CHP plants the index PES takes the mean value of about 25 %.

Figure 5.33 presents a small-scale CHP plant equipped with a piston gas engine.

In the derivation of the formula describing the index PES it was taken into account that the efficiency of transmitting heat and electricity is more favorable in a small-scale CHP plant than in their separate production and transmission.

For this purpose, the ratio of the efficiency of the transmission of electricity and heat produced separately to that of transmitting them in a small-scale CHP plant has been quoted:

$$\frac{\eta_{et}^{'}}{\eta_{et}} = \beta_1 \; \frac{\eta_{ht}^{'}}{\eta_{ht}} = \beta_2 \tag{5.95}$$

where

$\eta_{et}^{'}, \eta_{ht}^{'}$ efficiency of transmission concerning electricity and heat produced separately,

η_{et}, η_{ht} efficiency of transmission concerning electricity and heat produced in a small-scale CHP plant.

The relation concerning PES takes the following form:

$$\mathrm{PES} = 1 - \frac{1+\sigma}{\eta_{E\,CHP\,g\,G}\left(\frac{1}{\eta_{ref\,hp}\beta_2} + \frac{\sigma}{\eta_{ref\,pp}\beta_1}\right)} \tag{5.96}$$

where $\eta_{E\,CHP\,g\,G}$ denotes the gross efficiency of a small-scale CHP plant equipped with a piston gas engine.

As reference values and ratios β_1 and β_2 the following values have been assumed: $\eta_{ref\,pp} = 0,39;\ \eta_{ref\,hp};\ =\ 0,9;\ \beta_1 = \beta_2 = 0,95$

The results presented in Fig. 5.34 indicate that the realization of distributed cogeneration systems based on piston gas engines ensures the attainment of high values of the index PES.

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